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INVESTIGATION OF SHAFTING FAILURES AND ENGINE VIBRATION ON VESSELS OF THE *LOUISIANA* CLASS.

BY ERNEST N. JANSON AND LIEUTENANT COMMANDER J. O. RICHARDSON, U. S. N., MEMBERS.

PRELIMINARY DISCUSSION.

For some time past considerable trouble has been experienced in several of the *Louisiana* class of battleships from broken shafts together with an appreciable amount of engine and hull vibration at cruising speed.

The serious interferences generally caused by frequent shafting breakages on these ships and the stresses and inconvenience caused by vibration in engines and ship when running at cruising speed became a matter of urgent consideration. Therefore a Board was appointed and the U. S. S. *Minnesota* placed at its disposal for preliminary observations, followed by observations on the U. S. S. *Kansas*.

The question of shafting breakage on these ships has been

investigated mathematically as to designed strength from time to time and the physical condition of the material ascertained by photomicrographic and other tests. In no instance, however, could the cause of the breakages be traced directly to sources thus divulged, the shafting, according to the established rules used for the calculation of stress and dimensions, in each instance having possessed the customary allowance for safety, and, but for crystalline appearance of the fracture in certain cases, the material was found to conform fully to requirements as specified by the Government. It should be stated, however, that tensile tests made on the broken shafting of the U. S. S. *Connecticut* disclosed a diminution of the original strength from 92,845 pounds to 89,167 pounds per square inch, with a corresponding decrease in the modulus of elasticity for shear from about 11,850,000 pounds to 11,380,000 pounds. This is difficult to account for except possibly by assuming a condition of fatigue of the material having been brought about by continued severe stresses of a peculiar nature.

Investigations of the conditions which contribute to unbalance of the main engines were made by methods usually employed in ascertaining conditions conducive thereof. These calculations rendered evidence of only nominal unbalance of reciprocating forces, but of the existence of unbalanced moments of considerable magnitude due to reciprocating and rotating parts. Accordingly very good reasons presented themselves for assuming that said unbalanced moments would perhaps constitute, if not the only or even principal cause of vibration, at least a contributory one of substantial influence. Said unbalanced moments having thus been closely ascertained as to magnitude by calculation, their real influence upon creating vibration in engine and hull were, however, yet only relatively known, as, for instance, by comparison of effects found by similarly calculated unbalanced moments in other ships.

Based upon data obtained by experiments and research work performed by various eminent engineers on the subject of synchronous torsional shaft vibration, calculations were made

to attempt a determination of revolutions per minute corresponding to the *critical speed* of engine, or the speed where synchronism of forced and natural vibration of individual parts as, for instance, the shafting, occurs.

The propeller is sometimes the source of hull vibration, and vibration is then most perceptible in the stern. It is due to various causes, such as the water not having a free run to the propeller, to unbalance of blades or hub or a blade being out of true. Any of the foregoing causes with present accuracy in the making of propellers and excellence in the design of form of the hull precludes somewhat the cause being traced to these sources. Difference in the velocity of the water over the surface of the propeller due to " wake" may, however, be a disturbing factor in causing hull vibration, due to a momentary increase of pressure in passing the hull by the blade.

It seemed, however, from general observations made during the trials that the propellers were, on the whole, non-contributory in creating any of the peculiar vibration to which the engines here were subjected.

CONDITIONS UNDER WHICH TESTS WERE PERFORMED.

Before any tests or trials were undertaken on the *Minnesota,* re-alignment of the entire shafting with the ship water-borne had been performed. Thrust and bearing foundations, together with the supports of the bearing between I.P. and A.L.P. cylinders had also been strengthened.

Four-bladed propellers in place of former three-bladed ones had been fitted on the *Minnesota,* but the old three-bladed screws were used on the *Kansas.*

The piston valves were run on the *Minnesota* as designed with ordinary balance pistons attached. The *Kansas'* engines were fitted with Lovekin assistant cylinders.

Dock trials of the engines with propellers *disconnected* were run at varying speeds of from 30 to about 96 revolutions per minute.

Dock trials of engines with propellers *connected,* the engines

running in opposite directions at speeds varying between 30 and about 65 revolutions per minute.

Trials *under way* at sea, in deep water run at various speeds.

A pallograph with electro-magnetic contact on the line-shaft coupling and timed to register with the A.L.P. at top in upstroke was placed in different parts of the ship (3 positions) and pallograms obtained for determination of *hull vibration*..

A Denny-Edgecombe torsion meter was placed on the starboard line shaft for the purpose of securing records of the *torque* on the shaft at different speeds, especially those which establish the beginning and end of the *critical speed* period, together with what may be designated as the "revolutions of maximum vibrations."

A long-arm gage with a heavy bob weight suspended at the unsupported end was placed to span H.P. and I.P. valve-chest cover flanges to ascertain the amount of vertical vibration at the slip joint between the two pairs of cylinders.

No means of any kind had been introduced to secure a different engine balance from that existing due to the original design.

The purpose of running the engines uncoupled was to establish the effect of the *unbalanced forces* together with their *moments* on the engine in creating vibration. Operating the engines in this manner, the influence of the propeller is entirely eliminated as well as conditions due to the forces created by torque on the shaft influencing torsional vibration. In other words, the object was to ascertain to what degree the *unbalanced* condition of the engines may be responsible for vibration.

No special arrangements were made on the *Kansas* to secure improved working conditions during her vibration tests.

UNBALANCED FORCES.

Before proceeding further, it may be of interest to have an understanding of what constitutes unbalanced forces and moments in a reciprocating engine. Suppose, therefore, steam is

admitted to a cylinder, under or on top of the piston, and the shaft is so fixed that the engine can not revolve. Then the forces due to the steam pressure within the cylinder are perfectly balanced, the pressure upward being exactly the same as the pressure downward, the weight of reciprocating parts being borne by the ship. Now as soon as the engine begins to turn over, setting in motion its moving parts, reciprocating and rotating, a certain portion of the pressure acting on the piston is absorbed in creating energy to said moving parts and a difference, therefore, in the up-and-down forces ensues, dependent upon the amount of steam force thus absorbed; hence an unbalanced force, which is in magnitude equivalent to the inertia of the moving parts.

If for any one cylinder:

Force due to steam pressure $= P$,

Force due to inertia of reciprocating parts $= P_1$,

Force due to vertical component of the radial acceleration of rotating parts $= P_2$,

Force due to back pressure $= P_3$,

Force due to reciprocating weights $= W$,

Force creating torque (turning force) $= Q$,

Total pressure on bearings $= T$,

Distance between center of crank circle and the point where the connecting rod, or its extension, intersects a line drawn vertically to the center line of engine $= a$,

then, for the retarding portion of the stroke, the unbalanced force $=$

$$P - (P - P_1 - P_2) = P_1 + P_2,$$

and, for the accelerating portion the unbalanced force $=$

$$P - (P + P_1 + P_2) = -(P_1 + P_2).$$

The inertia force during a revolution is, of course, constantly changing in magnitude as a result of variation in angular advance of the crank as well as for varying revolutions by

change of angular velocity, while the weight of moving parts is a constant factor in creating inertia.

The turning force

$$Q = a\{P \pm P_1 \pm P_2 - P_3 \pm W\}$$

for the accelerating or retarding portion, as the case may be, the weights of moving parts to be added for down-stroke and subtracted in up-stroke. The pressure on the bearings will be

$$T = P \pm P_1 \pm P_2 - P_3 + W$$

for the accelerating or retarding portion, as the case may be, weight of moving parts having same value as in determining the turning force.

The weights of the reciprocating parts act, in a vertical engine, as a constant downward force and, therefore, have no influence whatever in creating unbalanced forces, which are due *only* to inertia.

UNBALANCED MOMENTS.

The unbalanced *forces* in an engine in themselves create no direct effect with respect to vibration, but do so only when considered in connection with the unbalanced *moments* set up by their action.

An unbalanced moment arises from an unbalanced force when acting on a certain lever. The moments dealt with are those produced by the inertia forces when referred to vertical and horizontal planes and, therefore, we have:

1. Vertical moments of reciprocating and rotating forces;
2. Horizontal moments of rotating forces.

The vertical moments are usually of a more disturbing effect, both owing to their greater magnitude and to the fact that the ship is stronger horizontally than it is vertically.

There are no unbalanced moments in a one-cylinder engine although an unbalanced vertical force exists, due to the rather small second-period forces as influenced by the length of the

connecting rod, but such moments arise, more or less pronouncedly, in all multiple-cylinder engines where the cylinders are placed side by side along the shaft. The unbalanced moment produced by such engines at any instant is the resultant moment of all the forces, acting on their individual levers with reference to a chosen point, at any one time.

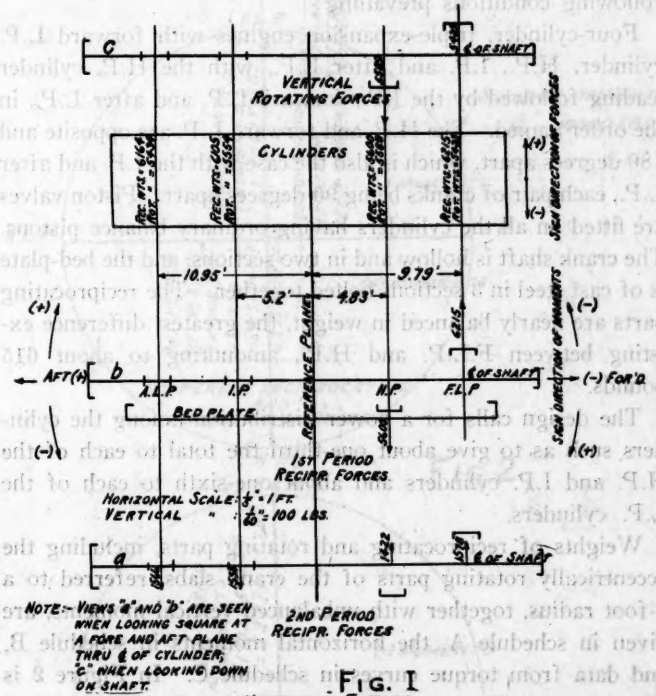


FIG. I
VERTICAL FORCES
OF
RECIPROCATING AND ROTATING PARTS.

As it should be hardly necessary to elaborate on this subject or to enter upon the general problem of unbalance and the means for the amelioration or prevention thereof, such having been done in various very able discussions by many engineers and investigators, only the most essential results as to magni-

tude of unbalance in the engines under consideration will be here cited, thereby enabling the reader to judge for himself what might seem as contributory causes of engine and hull vibration.

Referring to the sketch shown in Figure 1, which is representative of the engines as placed in the *Minnesota*, we find the following conditions prevailing:

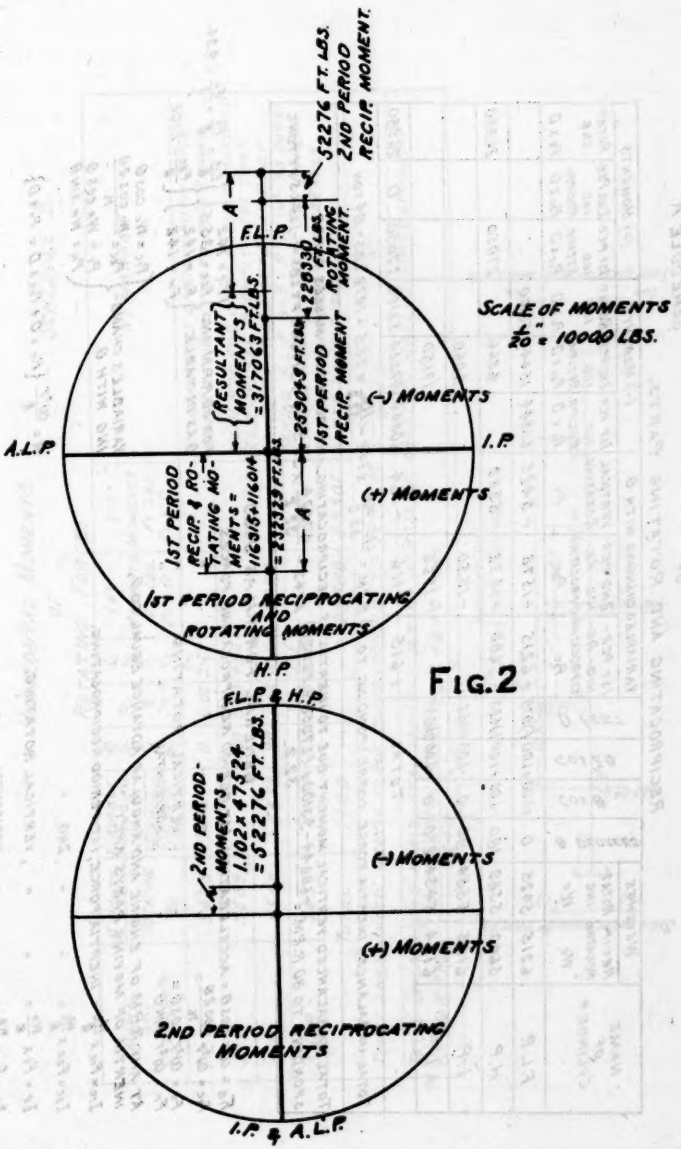
Four-cylinder, triple-expansion engines with forward L.P. cylinder, H.P., I.P. and after L.P., with the H.P. cylinder leading followed by the I.P., forward L.P. and after L.P., in the order named. The H.P. and forward L.P. are opposite and 180 degrees apart, which is also the case with the I.P. and after L.P., each pair of cranks being 90 degrees apart. Piston valves are fitted on all the cylinders having ordinary balance pistons. The crank shaft is hollow and in two sections, and the bed-plate is of cast steel in 3 sections bolted together. The reciprocating parts are nearly balanced in weight, the greatest difference existing between F.L.P. and H.P., amounting to about 615 pounds.

The design calls for a power distribution among the cylinders such as to give about one-third the total to each of the H.P. and I.P. cylinders and about one-sixth to each of the L.P. cylinders.

Weights of reciprocating and rotating parts, including the eccentrically rotating parts of the crank slabs, referred to a 2-foot radius, together with unbalanced vertical moments, are given in schedule A, the horizontal moments in schedule B, and data from torque curves in schedule C. In Figure 2 is shown graphically the solution contained in schedule A.

We find by reference to the schedules that, for 80 revolutions per minute of engine, which appears to be its critical low speed, the

Vertical unbalanced force = ...	.84	ton.
Vertical unbalanced moment = ...	145.5	ft. tons.
Horizontal unbalanced force = ..	.23	ton.
Horizontal unbalanced moment =	59.45	ft. tons.


52276 FT. LBS.
2ND PERIOD
RECIP. MOMENT.
A
F.L.P
228330 FT. LBS.
1ST PERIOD
ROTATING
MOMENT.
SCALE OF MOMENTS
1/20" = 10000 LBS.
{RESULTANT MOMENTS = 317063 FT.LBS.
253043 FT.LBS.
1ST PERIOD
RECIP. MOMENT
(-) MOMENTS
A.L.P
I.P
1ST PERIOD RECIP & ROTATING MOMENTS = 116315+116044 = 232333 FT.LBS.
A
(+) MOMENTS
1ST PERIOD RECIPROCATING AND ROTATING MOMENTS
H.P
FIG. 2
F.L.P & H.P
2ND PERIOD-MOMENTS = 1.102 x 47524 = 52276 FT. LBS.
(-) MOMENTS
(+) MOMENTS
2ND PERIOD RECIPROCATING MOMENTS
I.P & A.L.P

VERTICAL MOMENTS OF RECIPROCATING AND ROTATING PARTS.

SCHEDULE A.

NAME OF CYLINDER.	WEIGHTS		θ DEGREES	COS θ	COS 2θ	D FEET	VARIABLES CHANGING WITH θ			(-) MOMENTS			(+) MOMENTS		
	RECIPROCATING W_c	ROTATING W_R					1ST PER-100-RECIPROCATING P_{1c}	2ND PER-100-RECIPROCATING P_{2c}	VERTICAL ROTATING P_v	1ST PER 100. RECIPR. $P_{1c} \times D$	2ND PER 100. RECIPR. $P_{2c} \times D$	ROTATING $P_v \times D$	1ST PER 100. RECIPR. $P_{1c} \times D$	2ND PER 100. RECIPR. $P_{2c} \times D$	ROTATING $P_v \times D$
F.L.P.	6215	5425	0	+1.00	+1.00	(-)1879	+6215	+1578	+5425	60844	15448	53100			
H.P.	5600	5589	180	-1.00	+1.00	(+)1483	-5600	+1422	-5589		6866		27050		26980
I.P.	6105	5554	90	0	-1.00	(+)52		-1550			8060				
A.L.P.	6164	5434	270	0	-1.00	+11095		-1566			17150				
				TOTAL			+615	-116	-164	60844	47524	53100	27050	0	26980

TOTAL UNBALANCED INERTIA FORCE CORRESPONDING TO 80 R.P.M. $= \frac{615-164}{32.2} \times 142 - \frac{116}{32.2} \times 35.5 = 1872$ LBS. $= .84$ TON.

TOTAL UNBALANCED VERTICAL MOMENT DUE TO INERTIA OF RECIPROCATING AND ROTATING MASSES CORRESPONDING TO 80 R.P.M. $= \frac{-60844-53100+(27050+26980)}{32.2} \times 142 - \frac{47524}{32.2} \times 35.5 = (-)317063$ FT LBS. $= (-)141.5$ FT. TONS.

GENERAL FORMULAS USED

$F_{1c} = \omega^2 r \cos\theta$ = ACCELERATION, 1ST PERIOD RECIPROCATING FORCES.

$F_{2c} = \omega^2 r \frac{\cos 2\theta}{n}$ = " , 2ND " " "

$F_v = \omega^2 r \cos\theta$ = " , VERTICAL ROTATING "

$F_H = \omega^2 r \sin\theta$ = " , HORIZONTAL " "

AT ANY R.P.M. OF ENGINE AND ANGULAR ADVANCE EQUAL TO θ, INERTIA OF MOVING PARTS ARE:

$I_{1c} = F_{1c} \times \frac{W_c}{g}$ = INERTIA FORCE, 1ST PERIOD RECIPROCATING.

$I_{2c} = F_{2c} \times \frac{W_c}{g}$ = " " , 2ND " "

$I_v = F_v \times \frac{W_R}{g}$ = " " , VERTICAL ROTATING.

$I_H = F_H \times \frac{W_R}{g}$ = " " , HORIZONTAL "

FOR 80 R.P.M. AND θ AS IN TABLE $\begin{cases} F_{1c} = 142 \\ F_{2c} = 35.5 \\ F_v = 142 \\ F_H = 142 \end{cases}$ $\left\{ \begin{array}{l} \frac{F_{1c}}{F} = \frac{F_v}{F} = \frac{F_H}{F} = 4.34 \\ \frac{F_{2c}}{F} = 1.102 \end{array} \right.$

VARIABLES CHANGING WITH θ $\begin{cases} P_{1c} = W_c \cos\theta \\ P_{2c} = \frac{W_c \cos 2\theta}{n} \\ P_v = W_R \cos\theta \\ P_H = W_R \sin\theta \end{cases}$

$M = \frac{\omega^2 r}{g} \{P_{1c} \times D + P_{2c} \times D + P_v \times D\}$

$I_H = F_H \times \frac{W_R}{g}$ = " = " , HORIZONTAL "

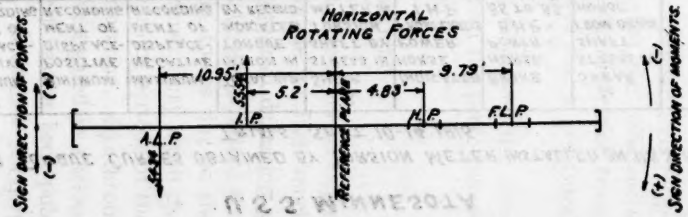
HORIZONTAL MOMENTS OF ROTATING FORCES

SCHEDULE B.

NAME OF CYLINDER	WEIGHTS ROTATING W_R	VARIABLE CHANGING WITH θ HORIZONTAL ROTATING F_H	θ DEGREES	SIN θ	D-FEET.	(+) MOMENTS FT. LBS. $F_H \times D$.	(−) MOMENTS FT. LBS $F_H \times D$	REMARKS
F.L.P.	5425		0	0.00	(−) 9.79			THE QUANTITIES GIVEN IN BOTH SCHEDULES A AND B REFER TO EACH ENGINE. MOMENTS FOR CRANK POSITION ASSUMED (F.L.P. UP) ARE SLIGHTLY LESS THAN WOULD BE THOSE DETERMINED BY TAKING A.L.P. AND I.P. IN UP- AN DOWN-STROKE RESPECTIVELY. THE DIFFERENCE HOWEVER WILL NOT GREATLY ALTER CONDITIONS.
H.P.	5689		180	0.00	(−) 4.83			
I.P.	5554	+5554	90	(+) 1.00	(+) 5.2	28880		
A.L.P.	5434	−5434	270	(−) 1.00	(+) 10.95		59501	
TOTAL		+120				28880	59501	

TOTAL UNBALANCED INERTIA FORCE CORRESPONDING TO 80 R.P.M. $= \frac{4.34 \times 1.20}{2240} = .23$ TON.

TOTAL UNBALANCED HORIZONTAL MOMENT DUE TO INERTIA OF ROTATING MASSES CORRESPONDING TO 80 R.P.M. $= \frac{(-59501 + 28880) \times 4.34}{2240} = -59.46$ FT. TONS.



U.S.S. MINNESOTA

SCHEDULE C

DATA FROM TORQUE CURVES OBTAINED BY TORSION METER INSTALLED ON U.S.S. MINNESOTA.
TRIALS - SEPT. 10-14, 1915.

R.P.M.	Y Maximum positive displacement of recording pen from 0-line.	Minimum positive displacement of recording pen from 0-line.	Maximum negative displacement of recording pen from 0-line.	Total variation in torque indicated by recording pen.	S Shear stress in shaft by torsion meter in pounds per sq. inch.	Indicated horse power from cards I.H.P.	Brake horse power = B.H.P. = .85 to .95 of I.H.P.	f_s Shear stress in shaft from brake horse power.	Remarks.
72	1.39"	.4"	0	.99"	4684				Torsion meter constant for stress in shaft 15½" outside and 9½" inside dia. = 3370.
74	1.42"	.2"	0	1.22"	4785				
77	2.62"	0	.57"	3.19"	8829				
78	3.0"	0	1.04"	4.04"	10110	2584	.88 x 2584 = 2273	3970	
80	3.38"	0	1.47"	4.85"	11390	2905	.85 x 2905 = 2556	4340	
82	3.02"	0	.97"	3.99"	10177				
84	2.88"	0	.68"	3.56"	9705	3345	.88 x 3345 = 2943	4760	
88	2.58"	0	.21"	2.79"	8694	3509	.89 x 3509 = 3123	4930	
94	NOT TAKEN	NOT TAKEN				4817	.90 x 4817 = 4335	6252	

S = STRESS PER SQ. IN.
Y = AS IN TABLE.
S = Y x 3370.

BY CALCULATION

$f_s = \frac{51 \times T \times 15.5}{15.5^4 - 9.5^4}$

$T = \frac{B.H.P \times 1.35 \times 63000}{R.P.M.}$

CONSTANT = $\frac{63000 \times 1.35 \times 5.1 \times 15.5}{49300} = 135.8$

$f_s = 135.8 \times \frac{B.H.P}{R.P.M.}$

The corresponding quantities for the rated full speed, or 123.5 r.p.m., are

Vertical unbalanced force = ...	2	tons.
Vertical unbalanced moment = .	346.3	ft. tons.
Horizontal unbalanced force = ..	.55	ton.
Horizontal unbalanced moment =	140.5	ft. tons.

We find from the foregoing that the unbalanced rotating forces and both first and second period of unbalanced reciprocating forces are not very large, but that the unbalanced moments are of considerable magnitude. Consideration being given to this fact, one would readily believe that a great deal of vibration might result therefrom.

The trials on the *Minnesota,* however, disproved this contention completely, as may be inferred from the performance during the trials with the engines uncoupled.

TORSIONAL VIBRATION OF SHAFTING.

The shafting, especially the crank shaft, due to its elasticity, is extremely sensitive to variations in torque, in fact, much more than generally is supposed. The torsional deflection is never less than the amount corresponding to steam forces acting in the cylinders together with inertia forces due to the moving masses of the engine. It is, however, very much exceeded in cases where synchronism occurs in the periods of the forced and natural periods of the shaft vibration. Synchronous torsional shaft vibration has been investigated in various ways by engineers, articles on the subject having been published from time to time. (See JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XIV, p. 563; XIV, p. 721; XXV, p. 563.)

While there has been a very strong supposition of this condition prevailing in the shafting of the class of ships in question, especially when running at cruising speed, there have been no means available for bringing out clear evidence of the ex-

istence thereof, and especially to establish the limits within which the vibrations actually were taking place.

Instruments or torsion meters of different designs have been used in connection with the measurements of the amplitudes of the oscillations corresponding to torsional vibration at any given number of revolutions of engines, the torsion meter used on the recent trials of the U. S. S. *Minnesota* and U. S. S. *Kansas* having been specially made by Messrs. Denny-Edgecombe, of Dumbarton, Scotland, for use in reciprocating-engine work.

In its operation a fairly accurate transposition of the torsional shaft deflection during each revolution is obtained, the torsional deflection for that part of the shafting to which the instrument is applied being shown on a paper record. The recording of the twist in the shafting is common to all torsion meters, those now used in turbine work, however, recording only maximum twist corresponding to any given power, while with this type of torsion meter a complete record is obtained for a continuous revolution.

In Figure 3 are reproduced diagrams taken with this meter on the U. S. S. *Minnesota* and in Figure 4 comparative curves of the U. S. S. *Minnesota* and U. S. S. *Kansas*.

In Figure 5 are shown three curves plotted directly from the curves shown in Figure 3, giving torsional displacement of the *Minnesota's* shaft.

In Figure X is given a good general view of the meter mounted ready for work.

Torsional displacement shown by curves is referred to a zero line, or a line of 0 torque, obtained by simply jacking the engine over when without load.

These torsion-meter records show that, at low speeds, the twist of the shaft is due solely to the power impulses delivered by the engine, and that at four points in each revolution when the engine is developing the maximum torque the shaft twists up the maximum amount and at the four points when the engine is developing the minimum torque the shaft untwists.

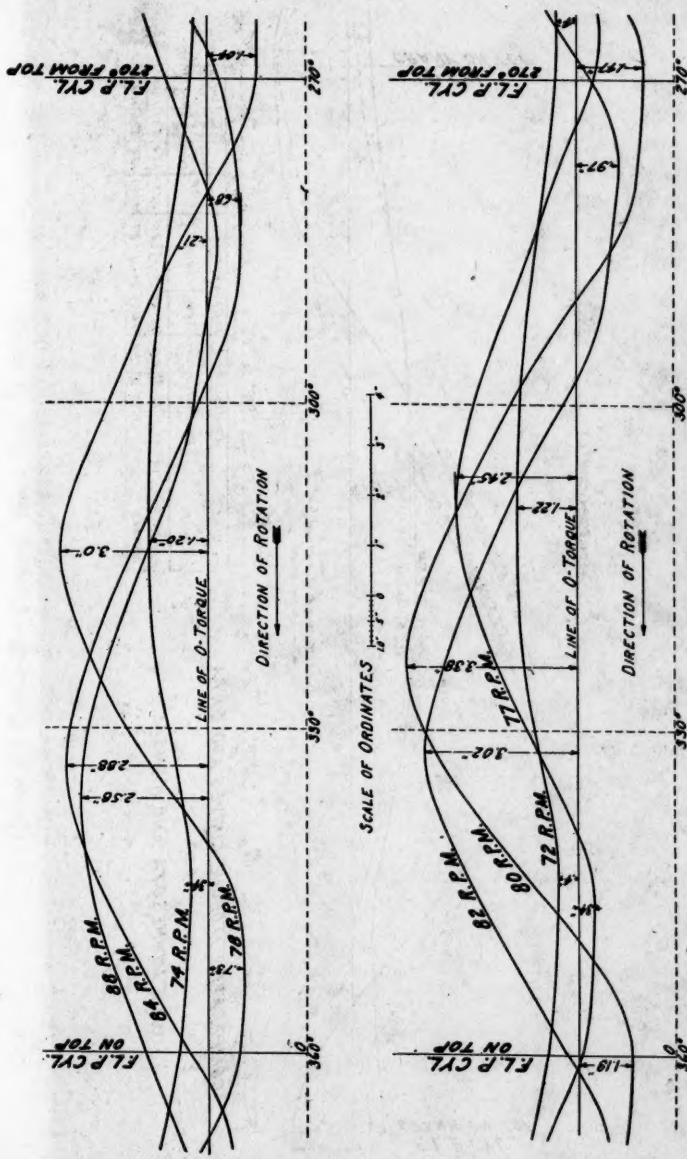
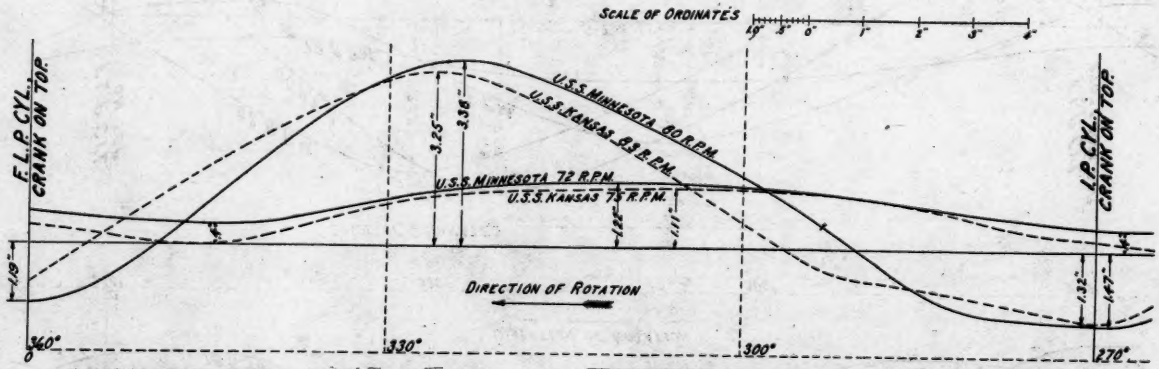


FIG. 3. U.S.S. MINNESOTA. TORQUE CURVES.



COMPARATIVE TORQUE CURVES AND DATA FOR U.S.S. MINNESOTA AND KANSAS.

FIG. 4

NAME OF SHIP	PROPELLERS				DRAFT ON TRIAL		R.P.M.	I.H.P.	SPEED IN KNOTS	DISPLACEMENT	SHEAR STRESS LBS. PER SQ. IN.
	NO. OF BLADES	DIAM.	PITCH	PROJ. SURFACE	AFT	FOR'D					
MINNESOTA	4	17'-3"	17'-9"	91.53	26'-10½"	25'-9½"	80	5950	13.4	339	11360
KANSAS	3	17'-3"	18'-0"	72.5	27'-7½"	25'-1½"	83	5150	13.5	325	10932

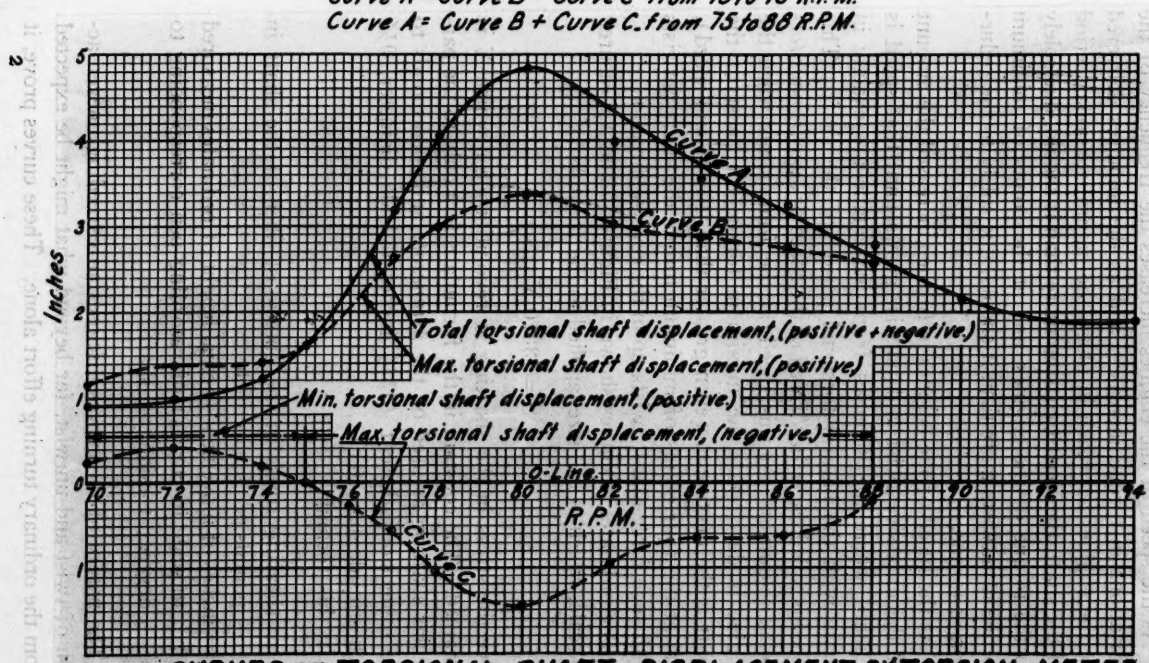


FIG. 5 CURVES OF TORSIONAL SHAFT DISPLACEMENT BY TORSION METER. ON U.S.S. Minnesota, Sept. 10-15, 1915

As the speed of the engines increases the frequency of the variations in torque increases, and as the impulses delivered by the engines approach in phase the natural torsional period of the shaft, the shaft twists up more than it should, due solely to the torque of the engines, and at the points of minimum torque it untwists more than it should, due solely to the reduction in torque.

As the speed of the engines is further increased the amount of the excessive twist and untwist grows until that speed is reached where the impulses from the engine are exactly in phase with the natural torsional period of the shaft. This speed is called the *critical speed* or the speed of *synchronous torsional vibration.* At this speed, at the four points when the engine is developing the maximum torque the twist of the shaft is the maximum (even greater than it is at full speed), and at the four points of minimum torque the shaft untwists with such force that it twists up in the reverse direction.

The ordinates in curves shown in Figures 3 and 4 measured to scale, which is full size, applied to the formula

$$S = 3370\ Y$$

give the stress in the shafting at revolutions for which Y is obtained. The constant in the formula refers only to the particular shafting as used on the *Minnesota* and corresponds to the dimensions thereof (15½ inches outside diameter, 9½ inches inside diameter).

In this formula

S = shearing stress in the outermost fiber of the shaft in pounds per square inch.

Y = inches of movement (actual height of ordinates measured to scale in curves) of recording pen corresponding to degrees torsion of shaft.

The nature of the curves discloses an exceeding uneven action of twist and untwist, far beyond what might be expected from the ordinary turning effort alone. These curves prove, it

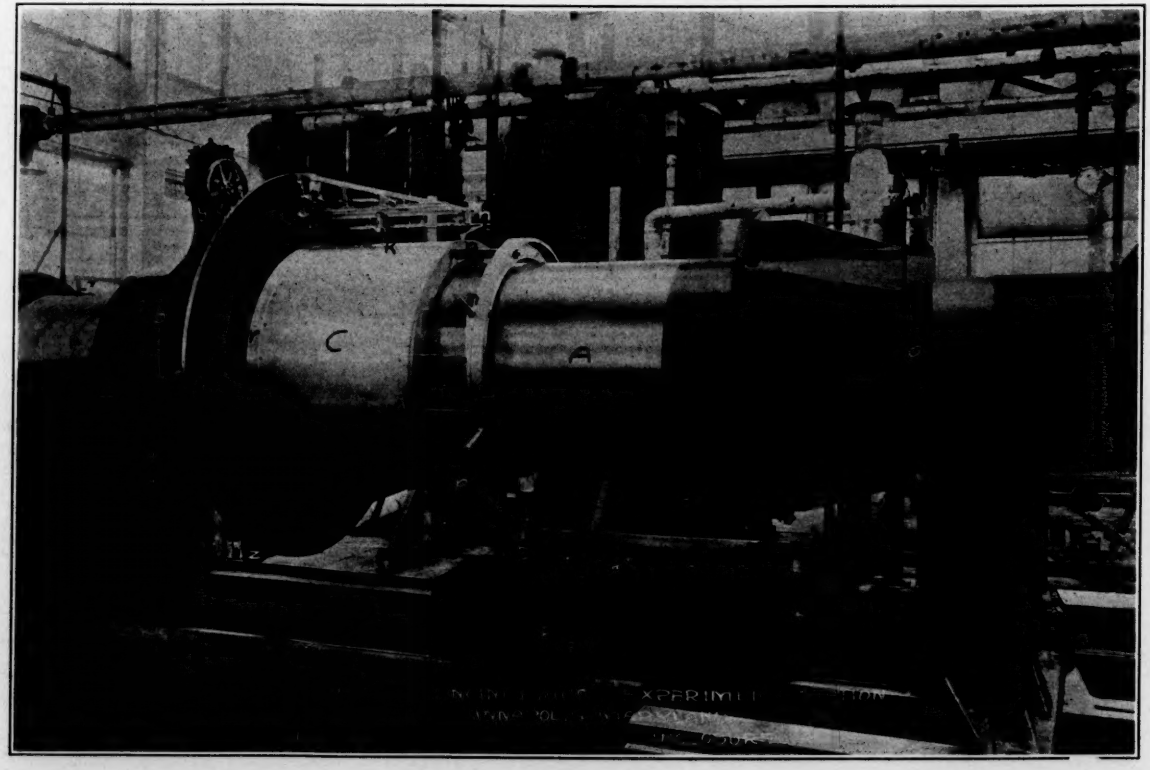


FIG. X.

seems conclusively, that there are forces at play in the shafting not anticipated and wholly unprovided for in figuring dimensions from stress.

PERFORMANCE OF ENGINES OF U. S. S. "MINNESOTA."

A. *Engines unconnected.*—The performance of the engines disconnected, in so far as vibration is concerned, at all speeds within the range used (and there is no reason to surmise a different performance at any other, higher, speed) was in every respect satisfactory, running smoothly, quietly and absolutely without any disturbance to hull or engines from vibration, a slight tremble only in columns and rods being discernible, due, no doubt, to slack pins and journals. The pallograph records taken during these runs showed slight vertical hull vibration or a little less than 1/16 inch as the extreme amplitude in the stern vibration vertically, but no horizontal vibration.

There was no appreciable hull vibration in the engine spaces or shaft alleys.

Torsionmeter records were, of course, not obtained during this trial, both owing to the position of the meter, it being located abaft the thrust shaft, on the unconnected part of the shaft, and to the fact that under this condition of operation practically no torque was transmitted by the shaft.

B. *Engines connected, dock trial.*—Indicator cards were taken on both engines, thus rendering complete records of power during simultaneous ahead and backing conditions.

Pallograms were taken and some torsion meter cards.

C. *Under way, at sea.*—(September 11, 12, 13, 14 and 15, 1915). The pallograph on this trial was placed in two different positions, at frame No. 57 amidships and away forward in the paint locker.

Torsion-meter records were taken from 69 revolutions up to 94 at appropriate intervals.

Main bearings gave trouble on account of heating. Thrust steady-bearings as well as spring bearings gave similar trouble, especially at or near the *critical speed* of the engine.

Interest centered around securing torsion-meter records for close intervals of speeds, and especially at speeds where the vibration was serious.

Torsion-meter records up to 72 revolutions per minute show no twist or torsional deflection of shafting beyond that due to known engine forces. Beyond this point the tendency is towards a more rapid rise, and at 78 revolutions the torsion-meter records show an abnormal condition to exist, the twist of the shaft having already, at this point, considerably exceeded that caused by the engine forces as given by established methods of determination.

At speeds from 78 to 82 revolutions, proportionate to the power developed, the records show a still more rapid increase in the shaft torsional deflection, reaching a maximum at about 80 to 81 revolutions. At this speed an entirely irreconcilable condition developed, consisting in an enormously increased torque in one direction, together with a complete reversal of same, and torque of considerable magnitude in the opposite direction, creating a to-and-fro motion, or twist and untwist, recurring with a frequency depending on the exact revolutions made per minute. For the ship in question synchronous torsional shaft vibration appears to begin at a speed of approximately 76 revolutions, reaching a maximum at about 80 to 81, and practically ceasing at about 86 revolutions per minute.

For higher speeds of revolution, or those beyond 86, a gradually normal condition is again attained. This was plainly observed as well as shown by the torsion-meter records. Thus at 94 revolutions we have approached a condition of torque corresponding to that which may be proved should exist as due to calculable engine forces.

ACTION OF ENGINES AND SHIP DURING "CRITICAL" SPEED OF REVOLUTIONS.

To effectually describe the action of engines and hull of the U. S. S. *Minnesota* during the "critical" speed performance of her engines is difficult owing to the impossibility properly to convey impressions stimulated by sound, vision and touch.

The sound produced in the engines is of the grinding, clattering kind heard when hard bodies move eccentrically against, or abruptly strike, one another. This noise undoubtedly comes from the sudden reversal in the torque of the shaft, whereby blows are delivered by pins, journals and eccentrics to stub ends, eccentric straps and bearings. A crackling, somewhat enervating, noise is developed in the ship structure extending to the vicinity above and below the engines and shaft-alleys. A note of a low pitch was audible in the inner bottom under the engines due in a measure to vibration of the hull at this place.

The athwartship vibratory motion of the eccentric rods created a blurred picture and the irregularity could be plainly observed. Instead of a smooth, oscillating motion, the rods moved with a jar which produced rattling and severe vibration, the intensity of which increased with the approach to critical speed where it reached its maximum, and then again became practically normal just above 84 to 86 revolutions of the engines.

With respect to the main bearings, especially those designated as No. 5, being overhung and located between I.P. and A.L.P. cylinders, a distinct movement could be observed, consisting of a partial screw motion together with an up-and-down motion of the ends.

There was practically no vertical motion of the cylinder units relatively one to the other at the expansion joint between the valve chests of the high and intermediate-pressure cylinders. The specially made gage showed by actual measurement this motion amounted only to .003 inch. This is, indeed, very small, considering each pair of cylinders is free to move up and down relatively to the other.

Spring and thrust bearings had an appreciable amount of movement both fore and aft and athwartship, and did not render the substantial, rigid support to the shafting which should be their purpose.

Some vibration in the columns was disclosed by the touch,

and engine-room floor plates, especially opposite I.P. cylinder, moved up and down. The crank shaft seemed to attain a slightly eccentric motion.

Other abnormal occurrences which developed may be summed up in the following:

1. Slacking off of nuts on main-shaft coupling bolts whereby the couplings would gradually get loose.
2. Heating of thrust-block steady-bearings.
3. Shaking loose of key and set screws in the wormwheel of main-engine jacking gear.
4. Vibration of inner and outer skin of ship below engines.

THE PERFORMANCE OF THE ENGINES OF THE U. S. S. "KANSAS."

The engines of the *Kansas* are practically the same as those of the *Minnesota,* both as to general arrangement and details, the foundations of bearings and bedplates, however, although not materially differing in design from the *Minnesota's,* may have benefitted by more substantial construction and somewhat better arrangement.

While the same characteristic features present themselves in the action of the engines as regards operation of detail parts, such being especially noticeable in the athwartship vibratory motion of the eccentric rods, the general effects produced by torsional vibration is considerably less, both as regards violence of action and disturbance to engine and hull.

Vibration of the hull as caused by the propeller seemed at times to exist in the *Kansas* in a more aggravated form than in the *Minnesota* under similar conditions.

On the whole, it may be stated, however, that the engines of the *Kansas* did operate under far less stress and generally perform under less disturbing conditions than did the *Minnesota's* engines. This is principally due to the shafting retaining more perfectly its alignment, thus sustaining lessened stress from bending. As a result thereof friction is somewhat reduced together with deleterious effects from reactions of shaft deflection.

CAUSE OF VIBRATION.

1. Synchronous torsional shaft vibration.

2. Structural weakness in shafting supports and engine foundations and of engine bed-plates.

Based upon certain calculations made in the Bureau of Steam Engineering for this class of ships at the time of more serious notice of vibration and shafting breakages, consideration was given to the probable existence of what has previously been described as "synchronous torsional shaft vibration."

Nearly all reciprocating engines no doubt at some certain speed develop more or less synchronous torsional shaft vibration, but not even nearly all give any serious trouble. The reason for this must be looked for as being not only that the critical speed does not lie within the range of revolutions used, but that the scantlings and construction of foundations and hull are such as to carry off the energy of the impulses created by engine vibration, thereby largely neutralizing its effects. While the *Kansas'* engines come well within reaching critical speed conditions, it is possible that due to certain details in her construction the effects of the vibration are greatly ameliorated.

In an analysis of the causes of engine vibration the question of engine balance enters as an important factor. While the engines in question do not possess "perfect balance," it may be stated that they are in a fair state of balance. Computations disclose the existence, as stated before, of unbalanced moments of apparently considerable magnitude, from both reciprocating and rotating forces, but the effect of said unbalanced moments seems insufficient to bring about vibration of noticeable amount in the engines or hull at any speed, as shown by their operation when disconnected. It is safe to conclude, therefore, that this cause is not contributory. Inherent weakness due to design of bed-plates, especially as regards No. 5 bearing, it is believed, aggravates the trouble.

During the performance of the machinery the working of

foundations for engines and bearings and the sensitiveness of the hull along the line of shafting, shown by changes caused by apparently insignificant disturbances, prompt the assumption that engine and bearing foundations are lacking in strength and rigidity and generally that the shafting is insufficiently supported.

SHAFT BREAKAGES.

Metals generally, and steel in particular, become fatigued under abnormal variation in stress, whereby the strength is greatly impaired. The shafting in a reciprocating marine engine is subject to such stress due to highly fluctuating torque and to bending moments of more or less importance.

From causes previously given there exists in this class of vessels an exceptionally high torsional stress in the shafting at a speed used when cruising. This, together with stresses caused by unduly large linear deflection, as a result of the shafting being insufficiently and weakly supported, subjects it to stresses beyond the limit of ordinary safety.

Should the metal have become fatigued (which is highly probable) after years of service under conditions alluded to, this condition, together with abnormal stresses, explains, in many cases at least, the probable cause of breakage. Crystallization of the metal has probably never occurred owing to the likelihood of the stress never having equaled the elastic limit.

REDUCING THE EFFECTS OF TORSIONAL VIBRATION.

The amplitude and period of the natural vibration depend upon the masses composing propeller and engine moving parts together with the force of elasticity of the shafting. Similarly the period of the forced vibration depends upon revolutions of engine and the number of variations per revolution, the amplitude being a function of the mean force and the variations.

The expression "variation" implies all the harmonic curves that the crank effort diagram may be resolved into by means of Fourier's Theorem (for the application of which the reader

is referred to " Calculus for Engineers" by Perry). The number of individual or natural vibrations may be found approximately by the following formula formerly cited in connection with the subject of torsional vibration.

$$N_E = \frac{30}{\pi \times R} \sqrt{\frac{G \times I_p (m + M)}{L_o \times m \times M}}, \quad . \quad . \quad . \quad (1).$$

Where

N_E = Natural period of vibration, or individual vibrations per minute.

R = Crank radius in inches.

G = Modulus of elasticity for shear of the shafting material. For the ships in question tests made on some of the broken shafting show that G here is approximately = 11,380,000 pounds.

I_p = Polar moment of inertia in inches.[4]

m = One-half the mass of reciprocating parts, all engine rotating parts including crank shaft journals and pins, crank slabs, 3/10 the weight of all connecting rods and 1/3 the weight of line shafting, counted from its center of gravity with this figured for engine and propeller masses for length between propeller and after engine coupling. Foregoing weights must all be referred to the crank radius.

M = Mass of propeller together with 25 per cent. of its mass in form of entrained water, all referred to crank radius.

L_0 = Length in inches of shaft of uniform diameter having same resultant elasticity as original shaft counted from center of propeller hub to midplane of engine. (For reduction of shafting see JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXV, Nov., 1913.)

When the natural period of vibration has been found from the foregoing formula, the critical number of revolutions of the engine are:

$$\text{Ist order} = \frac{N_E}{1};$$

$$\text{IId order} = \frac{N_E}{2};$$

$$\text{IIId order} = \frac{N_E}{3};$$

$$\text{IVth order} = \frac{N_E}{4}.$$

Usually only those of the III and IV order are within the limits of engine speeds used, and, as we shall see presently, correspond with two speeds of which the lower is extensively used in the ships with which we are concerned.

In about the manner the natural vibration in shafting is produced by an external oscillatory force created by the inertia of the moving masses, so is the forced vibration in the shafting produced by the steam forces acting in the cylinders, and while the amplitude is due to the magnitude of the movement of the creating forces, the resultant amplitude depends completely upon the proximity or nearness with which the movement of the oscillations per unit of time come together or synchronize. Thus if close together an enormous increase in the amplitude occurs, and if far apart only a small amplitude of vibration is produced, and if sufficiently far removed would approximately be due to the amplitude of either creating force only.

If, for instance, the forced frequency of a vibration is of the natural frequency in the following proportions, then the resultant amplitude will be as follows:

Forced Frequency / Natural Frequency.	Resultant Amplitude / Natural Vibration Amplitude.
.1	1.01
.5	1.33
.9	5.26
.99	50.25
1.00	∞

For $N_E = 324$ we have:

$$\frac{32.4}{324} = .1 \text{ corr. to } 8.1 \text{ revs.}$$

$$\frac{162}{324} = .5 \quad " \quad 40\tfrac{1}{2} \quad "$$

$$\frac{201.6}{324} = .9 \quad " \quad 50.4 \quad "$$

$$\frac{320.76}{324} = .99 \quad " \quad 80.19 \quad "$$

$$\frac{324}{324} = 1. \quad " \quad 81 \quad "$$

From the above the influence of approaching synchronism in forced and natural vibration is plainly shown. However, owing to the damping effect by increased frictional resistances the resultant amplitude of the vibration can never become infinite, but may be greatly intensified and therefore dangerous. The speed corresponding to this condition is, as stated before, known as the *critical speed* of the engine.

Herr L. Gümbel, in his excellent article on Torsional Vibrations of Shafts (Transactions of the Institute of Naval Architects, 1902), shows, by the aid of Fourier's Theorem in the form of an example, the resolution of the tangential force of an engine into a constant mean force and harmonic forces. There is also given in this article a formula for the angle of torsion of the two end sections of the shaft which, when experimentally verified, may be applied advantageously.

It is as follows:

$$\mu + \zeta = \frac{P(M+m)\sqrt{k^2 + M^2 w^2}}{k w^2 R m^2}, \quad . \quad . \quad . \quad (2)$$

where

P = tangential turning force in engine.

R, m and M = as before designated.

w = angular velocity of the natural period of vibration.

k = constant depending on propeller resistance.

1st Proposition to reduce effects of torsional vibration :

Increase diameter of shafting uniformly throughout and of a sufficient amount to reduce torsional deflection to a safe minimum.

If N_E represents period of natural vibration corresponding to I_p, and N_{E1} corresponding to I_{p1}, then $N_{E1} > N_E$ when $I_{p1} > I_p$; also if w represents angular velocity corresponding to N_E and w_1 to N_{E1}, then $w_1 < w$.

The revolutions of engine for synchronous vibration under these circumstances will be proportionate to N_E and N_{E1} and P and P_1 representing the tangential turning force in the engine corresponding to conditions representing N_E and N_{E1} will change approximately in a ratio of the cubes of the revolutions. With r and r_1 corresponding to N_E and N_{E1} we get approximately (r_1 being $> r$) $P_1 = P \left(\frac{r_1}{r}\right)^3$.

According to formulas given in Gümbel's article we have for the amplitude of the vibrations at the propeller end of the shaft

$$\beta = \frac{PM}{m\,k\,w\,R'} \quad \ldots\ldots\ldots \quad (3)$$

and for the engine end

$$a = \frac{PM\sqrt{k^2 + M^2 w^2}}{m^2\,k\,w^3\,r^2} \quad \ldots\ldots\ldots \quad (4)$$

Substituting P_1 for P in the above formulas, we find that amplitude of the vibrations for the propeller end becomes exaggerated in a larger proportion than that at the engine, both, however, have increased. But where an increase in the shaft diameter is made for the same transmission of power the torsional deflection is reduced in proportion to the increase in the polar moment of inertia and, by the greater rigidity gained, a corresponding resistance to vibration has been obtained. By enlarging the shaft the period will be increased approximately as the square root of the ratio. Thus if the shaft is made of dimensions such that its polar moment of inertia is doubled

the period will thereby approximately increase 42 per cent., or in the ratio of 1 to $\sqrt{2}$.

2nd Proposition to reduce effects of torsional vibration.

By the addition of mass to the rotating engine parts, or by an increase in the mass of moving parts of engine.

By changing the relative value of "*m*" and "M" the period as given by formula (1) will increase or decrease depending on the direction of the change. Thus, for instance, by changing the relative value of the two masses by adding to "*m*," or from $M = 1.7m$ to $M = 1.3m$, the period will be decreased approximately in the ratio of $\sqrt{1.6}$ to $\sqrt{1.8}$, which is 94 per cent. of the amount with the masses undisturbed.

From formula (2) we find that for any substantial addition to "*m*," even disregarding the influence upon P as previously explained, will theoretically materially reduce the angle of torsion of the two end sections of the shafting. Thus, by adding, for instance, a rotating engine mass such that, when referred to the crank radius, the engine mass is increased by about 25 per cent., a reduction of nearly 30 per cent. is shown in the angle of torsion with the addition of mass. This should prove important as effecting a proportionate reduction in stress in the shafting as compared to operation with the lighter weights.

3rd Proposition to reduce effects of torsional vibration.

To avoid revolutions of engines, for any considerable period of time, corresponding to speeds at which synchronous torsional shaft vibration occurs.

From the experiments performed on the ships in question, the conclusion may be drawn that even the small range of 6 to 8 revolutions, half below and have above the critical speed, may be considered a safe limit in establishing speed to be avoided.

CONSIDERATIONS IN DESIGN OF SHAFTING.

It is well always to remember that twist in shafting at the same time as it is proportional to stress it is also proportional to length, and that the surface displacement due to twist in-

creases with the radius or the distance from the center of any particular point in rotation. Therefore we find greatly magnified conditions of surface displacement existing in the shafting at such points as cranks, eccentrics and at the periphery of couplings, and the disturbance, in the form of violent engine vibration described as influenced by torsional shaft vibration, is especially pronounced in parts connecting at said points.

On the basis of foregoing considerations, therefore, it becomes very desirable, not to say absolutely necessary, to keep the spring or twist within very narrow limits, which can be done by a correspondingly low stress. Hence long shafts, such as we have in large marine installations, must be made much stronger and heavier than is necessary were the shafts only of short or nominal lengths.

That this practice is actually observed and scrupulously adhered to in important marine installations—not because of the rules established by either private shipping associations or by governmental bureaus, *none* of which provide for the observance or cognizance of the relation of *stress* to *length* in shafting, but simply because of a measure of protection—is shown by the shear stresses which have been figured for the shafting of the following ships.

		lbs. per sq. in.
S. S. *Aquitania*	(Turbine)	4,160
S. S. *Imperator*	(Turbine)	5,850
S. S. *Britanic*	(Reciprocating)	5,950
S. S. *Britanic*	(Turbine)	4,860
U. S. S. *Minnesota*	(Reciprocating)	12,500

Furthermore, it is generally recognized as permissible and good practice to design the different parts of the shafting with reciprocating engines to conform with the stresses derived from combined torque and bending. These stresses, however, vary considerably for the different parts of the shafting owing to the great difference in bending moments, and hence result in establishing different diameters for propeller shafts, crank shafts,

and line shafts, the latter generally coming out much smaller than the others. This method, however, in view of stresses brought about by torsional vibration, must be considered impracticable and, at least in individual cases, is dangerous of application. When so designed, the unequal spring or twist throughout the length of shafting results in highly aggravated torsional displacement in other parts of the shafting to which the twist is transmitted, as, for instance, in the crank shaft when the intermediate line shafting is reduced.

CONCLUSION.

What has been previously described as constituting *synchronous torsional shaft vibration,* together with its effect upon operation of the engines and the creating of disturbing hull vibrations, is largely due to existing relation in arrangement and detail of machinery and hull. Such arrangement and detail are affected by location of engine with reference to propeller, the diameter and length of shafting, relative weight of engine moving parts and propeller, the elasticity of the shafting and by the revolutions of the engines. The engine vibrations, being primarily set up by forces due to intensified to-and-fro shaft vibrations, are only slightly dampened as a result of the lightness of engine structure and scantlings composing the hull under engines and shaft bearings, together with what appears somewhat indifferently located supports under same. Figures 6 and 7 illustrate in an elementary way relative positions of engine and hull.

In new designs of propelling machinery fairly satisfactory arrangements of details may be provided to suppress and perhaps neutralize the powers influencing synchronous torsional shaft vibrations, but in the event of the trouble existing in the completed ship, as in the ships of the class under consideration, the changes necessary are in the great majority of instances exceedingly difficult of accomplishment. Of the means available, as previously alluded to, we find the following as perhaps most expeditious:

1. Change in speed at which cruising is customarily performed.

2. Increase in diameter of shafting.

3. Adding to the rotating masses of the engine.

What is implied in the first clause affects considerably the strategical value of the ship and for that reason is not here considered, while both of the other suggestions are of mechanical importance and may be considered from such viewpoint. Increasing the diameter of the shafting changes its natural period of vibration and therefore removes the synchronous period from the present critical speed to one that is infrequently used in ordinary operation. Moreover, increased safety and lessened torsional vibration is secured by reduced primary stresses through increased strength.

If, for practical reasons, it should be impossible to enlarge such parts as propeller shafting and crank shafting, it seems some advantage would accrue by enlarging such parts of the line shafting as now are smaller than the parts mentioned, thereby limiting the amount of torsional displacement transmitted to the crank shaft through the line shafting.

Adding to the rotating masses of the engine, whereby a reduction of both the period of vibration and the angle of torsion is obtained, offers a solution not so readily applicable but theoretically promising. The most convenient and simplest form to do this is found in the form of a fly-wheel secured at a suitable place on the engine shaft abaft and near the engine. To accomplish fairly effective results by this method, the necessity arises to incorporate in the fly-wheel considerable weight and large diameter, and, therefore, the occupation of valuable space always difficult to spare in ship engine rooms, besides an additional heavy pressure on the bearings.

As the greatest importance is necessarily attached to a satisfactory solution of this problem involving excessive vibration together with aggravated stresses in engine as well as ship, a systematic procedure of experiments must be resorted to.

That the conditions prevalent on the ships in question are

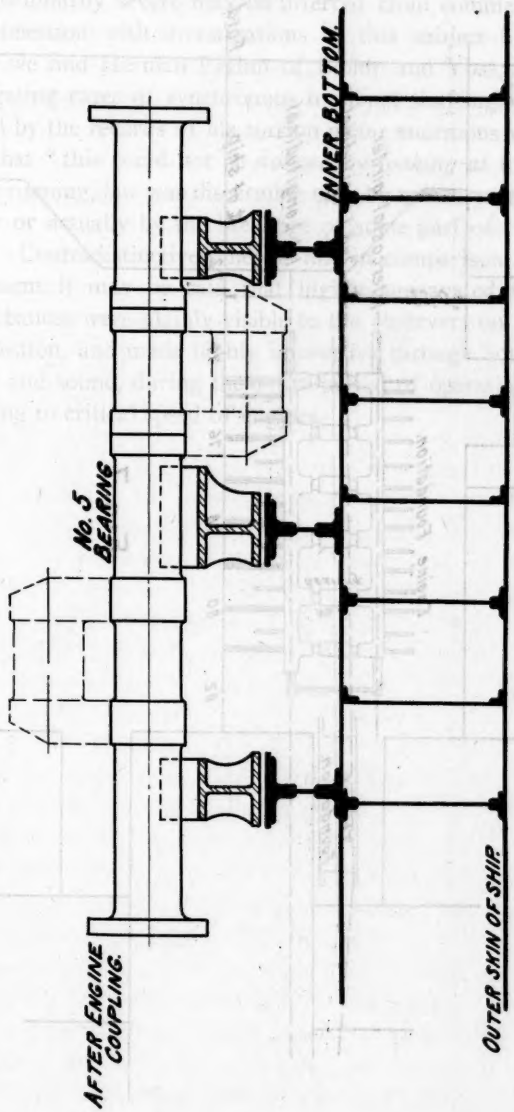

INNER BOTTOM.
No. 5 BEARING
AFTER ENGINE COUPLING.
OUTER SKIN OF SHIP.

FIG. 6

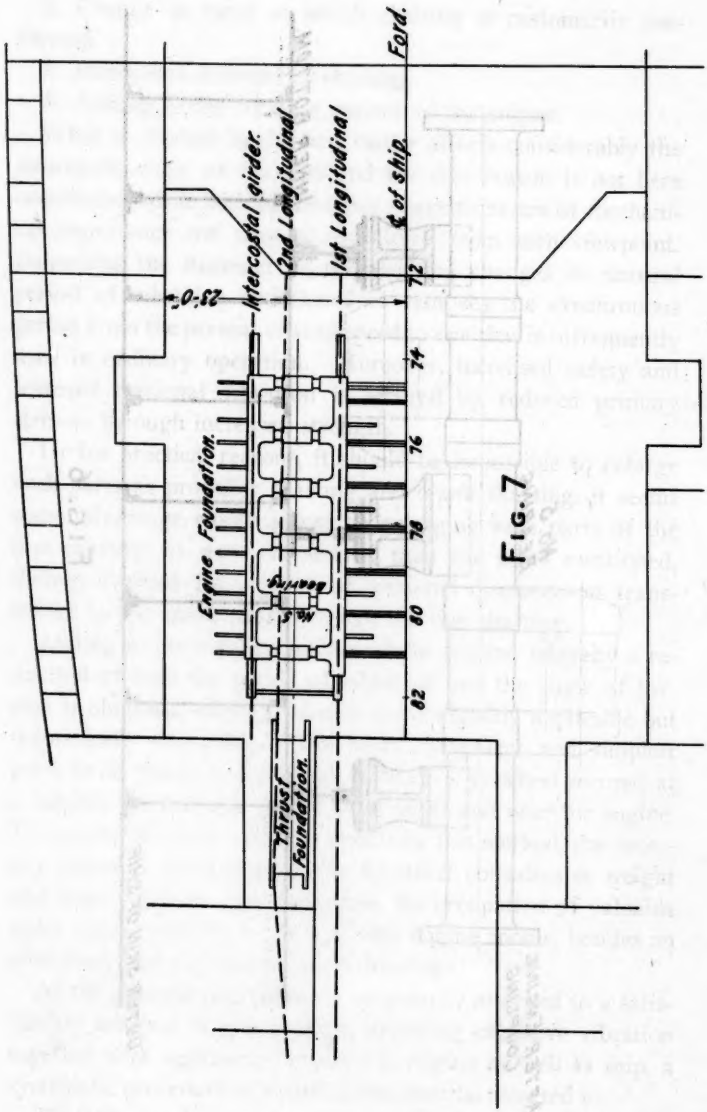

Fore.
Intercostal girder.
2nd Longitudinal
1st Longitudinal
℄ of ship.
Engine Foundation.
Thrust Foundation.
70
72
74
76
78
80
82

FIG. 7

extraordinarily severe may be inferred from comments made in connection with investigations of this subject by others. Thus we find Herman Frahm of Blohm and Voss, while investigating cases of synchronous torsional shafting vibration, found by the records of his torsion meter enormous vibration, and that "this could not be *noticed* by *looking at* the engine while running, but was discernible only by measurement of the meter or actually by the breakage of some part of the shafting." Contradistinctively hereto and in comparison with this statement it may be said that highly aggravated vibratory disturbances were plainly visible to the observers on the ships in question, and made highly impressive through both vision, touch and sound, during the entire period of operation corresponding to critical speed of engines.

DIRECTIONS FOR USING THE GAS ANALYSIS APPARATUS.

By Lieutenant-Commander W. T. Conn, Jr., U. S. Navy, Member.

The reasons for using the gas analysis apparatus and the benefits which may be derived from its use have been widely discussed by various authors, and this information is so generally accessible to the service that it seems necessary here simply to mention the general purposes of the use of such an outfit on board ship. At the outset the use of the gas analysis is to determine what faults, if any, exist in the methods of firing and operating the boilers and to enable the personnel to make an intelligent diagnosis of the combustion toward discovery of the sources of loss in efficiency and applying the proper remedies to stop this loss. This having been accomplished, the continued use of the gas analysis assists in maintaining a high boiler efficiency by indicating at once any reduction in economy, and the cause thereof. The analysis of flue gases, or products of combustion, is the separation and measurement, quantitatively, of the several constituents of the gaseous mixture, which for practical purposes may be regarded as carbon dioxide (CO_2), oxygen (O) and carbon monoxide (CO). The operation consists of drawing into a measuring burette a given volume of gas. This volume of gas is then passed successively through various chemical reagents or absorbents to remove the different constituents. After transferring the gas to each absorbent it is brought back to the measuring burette and the decrease of volume measured, thus indicating the percentage that the constituent removed by absorption bore to the original volume.

The details connected with the mixture of the chemical absorbents, the preparation of the outfit for use and the method of collecting the gas and analyzing it are not generally available for ready reference. The object of this article, therefore, is to present to the service a compilation of such information which may be of some aid to those engaged in the use of the apparatus, which seems lately to have received a new impetus.

There are many forms of the gas analysis apparatus on the market, and the Bureau of Steam Engineering has supplied a number of the various makes to different vessels. It is impracticable to include a detailed description of all makes within the scope of this paper, nor is it necessary, since all of the proprietary apparatus are simply modifications of the Orsat. The apparatus known as the *Orsat-Muencke* is fairly representative and a description of it will, with slight modifications, be applicable to all such instruments.

COLLECTION OF THE SAMPLE.

The collection of the sample is of equal, if not greater, importance than the actual analysis, for unless the sample obtained is representative of the average condition of the gases, the analysis will be worthless. Since the various constituents are constantly fluctuating it will be of little or no benefit to take "grab" samples of the gas at stated intervals, the reason being that such "grab" samples would show only the actual condition existing at that moment, and possibly bear no relation whatever to the average furnace conditions.

To insure that the sample of gas is truly representative of the *average* furnace conditions, instruments have been devised to draw a continuous stream of gas into a receptacle at a uniform rate during whatever period of time may be desirable, and it should be emphasized that unless the sample be obtained in this manner the time required to make the analysis is time thrown away.

The sample of gas for analysis should be taken from a point in the center of the gas current in the last pass of the boiler

after the gases leave the heating surface, and in all cases before reaching the damper. The gas is obtained by inserting through the boiler casing into the uptake a piece of ordinary ½-inch iron pipe of sufficient length so that the open end of the pipe is in the center of the current of gases; the other end of the pipe is bushed down to connect to the gas collector. Consensus of opinion appears to be that this open-end sampler is to be preferred to a perforated tube, that is, a pipe extending across the boiler, with both ends closed, and having a series of holes drilled into it at intervals along its length.

The gas-collecting bottles may be connected to the sampling pipe as shown in Fig. 1, a diagrammatic sketch of the collecting apparatus devised by the Bureau of Mines. Such a device may be readily made and installed by the ship's force. (A) is a regulating valve connecting to water supply to operate the ejector (C). The ejector, to create a flow of gas from the uptake, consists of a ¼-inch brass nipple, shaded in the figure, which is made of solid stock about 1½ inches long and drilled with a 1/16-inch axial hole; this is threaded and screwed into the tee (C). Into the other end of the tee is screwed the nipple (D), ¼-inch by 12 inches, forming the throat of the ejector, which is attached by a reducer to the ½-inch discharge pipe (E). The ejector is connected to the water-regulating valve (A) by the union (B). The gas-supply pipe (G) should have a downward lead from the uptake to the ejector, and is attached to the ejector and water-collecting bottles (L) as shown. (F) and (F^1) are ¼-inch brass cocks. (F) being used to admit gas to the collector. (F^1) may be connected to a draft gauge to indicate the suction created by the ejector, but need not be used unless desired. The connections between cock (F) and water bottles (L) are made of ¼-inch rubber tubing (H) and glass tubing (K) about ¼-inch inside diameter. Pinch cocks (I) are provided to regulate the intake of gas into the upper bottle (L) and chiefly to regulate the flow of water from the upper bottle (L) to the lower one.

To collect a sample of gas the upper bottle (L) is completely filled with water and the lower bottle is emptied. All pinch cocks (I) and cock (F) are closed, and the lower bottle is placed about 4 feet below the upper bottle. Open regulating valve (A) and start water through the ejector; allow this to run for a few minutes to create a suction of gas. Then open cock (F) and pinch cocks (I); the flow of water from the

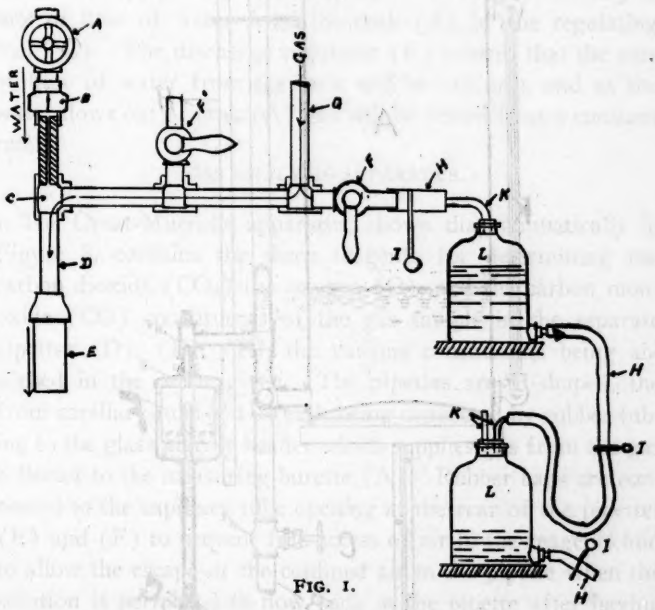
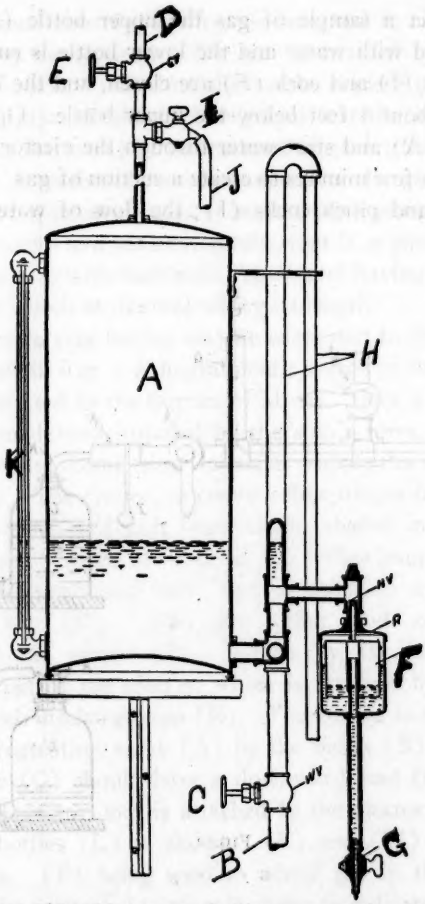


FIG. 1.

upper bottle to the lower will draw gas into the upper bottle, and this can be regulated to any desired rate by adjusting the pinch cock (I) on the rubber tubing connecting the two bottles. Air in the lower bottle escapes through the open glass tubing (K) inserted through the stopper.

The Hays automatic gas collector, Fig. 2, is a more permanent apparatus, and as it has been supplied, in the past, to a number of vessels a brief description is given.



Gas Collector, with Water Flow Regulator

FIG. 2.

(A) is the tank for collecting the gas and is connected to a source of cold-water supply by pipe (B). Admission of water to (A) is regulated by valve (C). To assemble and operate the collector, open (C) and admit water until the gage (K) shows tank to be about half full. Through opening at top of

tank pour in about one pint of lubricating oil and screw in pipe connections. The oil forms a seal between top surface of water and the gas which is to be collected in the tank, and prevents absorption of CO_2 by the water. The gas-sampler pipe in the uptake of the boiler is connected to (D). Close valve (E), open cock (I) and valve (C), and admit water to tank until overflow occurs through pipe (H). When this happens, shut off water supply, close cock (I), open valve (E) and adjust rate of flow of water from the tank (A) by the regulating cock (G). The discharge regulator (F) insures that the rate of flow of water from the tank will be uniform, and as the water flows out of tank (A) gas will be drawn in at a constant rate.

GAS ANALYSIS APPARATUS.

The Orsat-Muencke apparatus, shown diagrammatically in Figure 3, contains the three reagents for determining the carbon dioxide (CO_2) the oxygen (O), and the carbon monoxide (CO) constituents of the gas sample in the separate pipettes (D), (E), (F), the various constituents being absorbed in the order given. The pipettes are U-shaped, the front capillary-tube end of each being connected by rubber tubing to the glass pipe or header which supplies gas from the gas collector to the measuring burette (A). Rubber bags are connected to the capillary tube opening at the rear of the pipettes (E) and (F) to prevent free access of air to the reagents and to allow the escape of the confined air in the pipette when the solution is permitted to flow back in the pipette after having been used. These bags must be airtight and, after some period of use, require renewal. The front part of each pipette is filled with a bundle of glass tubes in order to increase the absorption surface of the reagent. In addition, the glass tubes in pipette (F) contain curved copper wires to maintain the cuprous chloride solution at a constant strength.

The gas-measuring burette (A) is a glass vessel graduated into 100 units, and has a volume of 100 c.c. It is surrounded by a water jacket to prevent sudden changes in temperature

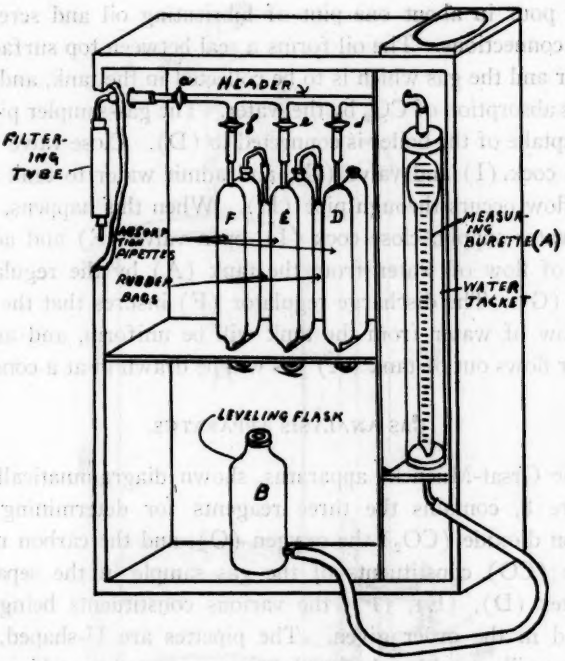


FIG. 3.

while analysis is being made. The burette (A) is connected by rubber tubing at the top to the header which supplies gas from the collector. The other end of the header is attached, by rubber tubing, to a glass tube fastened to the upper left-hand side of the case; this tube is filled with glass wool to serve as a filter for purifying the gas before it is allowed to pass into the apparatus.

The header, to which the measuring burette and pipettes are attached by rubber tubing, is a glass tube, fitted with a one-way glass stop cock (*d*), (*e*), (*f*), between the header and each pipette, and at the left-hand end there is a 3-way glass stop cock (*a*) to admit gas to the measuring burette or shut it off, and also to establish communication between the gas burette and the atmosphere. The glass stop cocks should be lightly

lubricated with vaseline. Care must be observed that all rubber tubing connections are gastight.

The leveling flask (B), which is filled with water, is attached by rubber tubing to the bottom of measuring burette (A). By raising (B) and opening stop cock (*a*) to the atmosphere, air in burette (A) is displaced by water which is allowed to rise to the uppermost mark on the burette.

SOLUTIONS.

Pipette (D) is used for the absorption and determination of CO_2. It contains a solution of caustic potash. A stock of this solution should be prepared by dissolving slowly one part, by weight, of caustic potash, purified sticks (not by alcohol), in two parts, by weight, of distilled water. A sufficient quantity of this caustic potash solution should be poured into the pipette so that when the solution is drawn entirely into the front leg of the pipette (the proper position for the solution when the pipette is ready for use), the front leg is filled up to the mark on the capillary-tube end, and the rear part is empty. The solution is drawn up into the front leg of the pipette (D) as follows: Stop-cock (*a*) is opened to atmosphere. Stop-cocks (*d*) (*e*) (*f*) are closed. Burette (A) is filled with water to its small neck by raising leveling flask (B). Stop-cock (*a*) is now closed, stop-cock (*d*) opened, and the solution in pipette (D) is drawn into the front leg by lowering (B). When the solution has reached the mark on the capillary stem of pipette (D), close stop-cock (*d*). Care must be taken not to draw the caustic potash solution into the stop-cock (*d*); the solution rises slowly as long as it remains in the large body of the pipette, but upon reaching the small capillary tube the rise is rapid. This rapid rise can best be checked by pinching with the fingers the rubber tubing connecting (A) and (B) and thus throttling the flow of water into the leveling flask. One c.c. of *caustic potash* solution will absorb about 40 c.c. of CO_2.

Pipette (E) is for the absorption of oxygen, the reagent

being a solution of pyrogallic acid. The solution is prepared by dissolving 20 grams of *pyrogallic acid* in 45 c.c. of distilled water; this is poured into pipette (E) and a sufficient amount of the *caustic potash* solution, used in pipette (D), added to bring the solution in pipette (E) to the same level as the solution in pipette (D); *i. e.*, both legs of the pipette are about one-half full. The rubber bag should be immediately attached to the capillary tube of the rear leg of pipette (E) to prevent the reagent from absorbing oxygen from the air. Just as in the case of (D), before using (E) for an analysis the solution must be drawn up to the mark on the capillary tube of the front leg of the pipette (E) and held there by closing the stop-cock (*e*). One c.c. of the pyrogallic solution will absorb about 13 c.c. of oxygen, but after some use the speed of absorption of oxygen by the solution is reduced, and when the rate of absorption becomes slow the solution should be renewed.

Pipette (F) contains an *ammoniacal cuprous chloride* solution to absorb the CO. A stock solution should be made up and kept for use in a tightly stopped bottle. The stock solution is prepared as follows:—Dissolve 250 grams *ammonium chloride* in 750 c.c. of distilled water; when the ammonium chloride is dissolved, add 200 grams of *commercial cuprous chloride*. This stock solution must be further treated for use in the pipette (F) by adding *one volume* of *ammonium hydroxide* (ammonia water, sp. gr. 91) to every *three volumes* of the stock solution. This final mixture should be poured immediately into the pipette (F), and the rubber bag attached to the capillary-tube end to prevent exposure of the solution to the air. One c.c. of the cuprous chloride solution will absorb about 16 c.c. of CO, but when determining small volumes or percentages of CO, a comparatively fresh solution should be used. As in the other pipettes, the solution must be drawn up to the mark on the capillary end of the front leg of the pipette before beginning an analysis.

MAKING THE ANALYSIS.

The pipettes being in readiness, that is, with the solutions drawn up to the reference mark on front leg about half way between stop cock and body of pipette, with stop cocks (*d*) (*e*) (*f*) closed and (*a*) open to source of supply, gas may be drawn into the measuring burette (A) for analysis. If the gas collector is in use (see Fig. 2), attach the rubber tubing (J) to the filtering tube of the gas-analysis outfit. Close valve (E), open cock (I) and turn on water by opening valve (C). Gas will be driven from the collector tank into burette (A), Fig. 3. Allow the gas to bubble through the leveling bottle (B) for a few moments, then close stop cock (*a*) and disconnect the outfit from collector tank. If the gas is not under pressure so that it can be forced into the Burette (A), Fig. 3, it is drawn in as follows:—Stop cocks (*d*), (*e*), (*f*) are closed as before and stop cock (*a*) opened to atmosphere; burette (A) is filled with water by raising leveling flask (B), care being taken that the water does not pass into the header. The header is connected to the gas supply, such as the upper collecting bottle (L), Fig. 1, stop cock (*a*) is opened to the gas supply and gas is drawn into the burette (A) by lowering (B). This sample of gas is then discharged from burette (A) by opening 3-way stop cock (*a*) to atmosphere and raising leveling flask (B). Several samples should be drawn into burette (A) and discarded before starting the analysis to insure that a representative sample of the gas to be analyzed has been obtained.

Particular care must be observed that exactly 100 c.c. of gas is taken into burette (A) and that this sample is under atmospheric pressure. This is done by drawing a sample of gas into burette (A) and closing the 3-way stop cock (*a*). The gas is placed under a slight pressure by raising leveling flask (B) until the water is somewhat above the zero mark on burette (A); pinch the rubber tubing connecting (A) and (B) between the fingers of left hand and lower the leveling

flask (B) below the lower end of (A). The pressure exerted by the left hand on the rubber tubing is gradually released and water in (A) allowed to fall until the bottom of the meniscus, or curve of the water, is exactly at the zero mark on (A). It is held at this point by pinching the rubber tubing, and the 3-way stop cock (*a*) is then momentarily opened to the atmosphere and again closed. The pressure of the fingers on the tubing is then released, and the leveling flask (B) is raised until the level of water in (A) and (B) is in the same horizontal plane passing through the zero mark on (A). It is desirable always to read from the bottom of the meniscus.

CARBON DIOXIDE (CO_2) MEASUREMENT.

The gas in burette (A) is now passed into pipette (D) for the absorption of the CO_2 constituent, by opening stop cock (*d*) and raising leveling flask (B). When burette (A) is filled to the top with water and the 100 c.c. of gas has been forced into pipette (D), the leveling flask (B) is slowly lowered and the gas is drawn back into burette (A). This operation is repeated three or four times, precaution being observed that the water does not pass into the header when filling burette (A), and that the solution in pipette (D) is always drawn up to the same reference mark on the capillary-tube end and is not allowed to enter the stop cock (*d*).

When the absorption is complete, bring the water level in burette (A) and leveling flask (B) to the same horizontal plane. One hundred (100) c.c. minus the amount of gas remaining in burette (A), as measured by the meniscus of water in (A), indicates the percentage volume of CO_2 absorbed by the caustic potash solution. Or, if the direct reading on the graduate of burette (A) is 12, then 12 per cent. of the original volume of the sample of gas has been absorbed by the reagent, and this is the percentage of CO_2 present. In order to be sure that all of the CO_2 has been absorbed the test should be repeated until a constant reading is obtained in burette (A).

OXYGEN (O) MEASUREMENT.

After the absorption of the CO_2, and the gas returned to burette (A), stop cock (*d*) is closed, stop cock (*e*) opened, and the operation of moving the gas above described is repeated, the gas being forced into pipette (E) for the absorption of the oxygen. Oxygen is not absorbed as rapidly as CO_2, and so the gas must be passed into the pipette (E) a greater number of times than was done in pipette (D). As before, to insure complete absorption of the oxygen, tests should be repeated until last two readings agree. When this has been accomplished, the volume of gas unabsorbed by the caustic potash minus the volume of gas now remaining indicates the percentage volume of oxygen which has been absorbed by the pyrogallic acid. Or, if the CO_2 reading was 12, and the reading on the graduate is now 16, then the oxygen absorbed occupied 16 minus 12, or 4 per cent. of the original volume.

CARBON MONOXIDE (CO) MEASUREMENT.

Stop cock (*e*) is now closed and stop cock (*f*) is opened and leveling flask (B) raised. The gas is forced into pipette (F), and the operation of absorbing the CO constituent continued, until the last two successive readings agree, showing that the CO absorption by the cuprous chloride solution is complete. The reading obtained after the oxygen absorption, minus the reading now shown on the scale of burette (A), gives the percentage volume of CO in the original sample. The remainder is generally regarded as nitrogen, although hydrogen is usually also present.

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STRAIGHT MINERAL OIL VERSUS COMPOUNDED OIL FOR THE BEARINGS OF MARINE ENGINES NOT HAVING FORCED LUBRICATION.

By Lieutenant (J. G.), J. L. Kauffman, U. S. N., Member.

The purpose of this article is to compare the relative advantages and disadvantages of straight mineral oils with those of compounded oils when used to lubricate the bearings of marine reciprocating engines that do *not* have forced lubrication.

It is realized that a great deal of marine machinery is lubricated by forced-feed methods, but a study of the results of the tests given in this article may enable the operators of the older types of marine engines to materially reduce the cost of the oil used.

For a great many years after the steam engine came into use for marine propulsion, all the bearings were lubricated by fixed oils, that is, some form of animal or vegetable oils, such as lard, tallow, sperm or olive. These oils could be obtained very easily and cheaply at first, and with the heavy, slow-moving machinery then used gave excellent results. As the number of engines increased and the amount of oil required became larger, the availability of the most favored of these, olive oil, decreased materially, thereby making the price excessive for the economical running of the machinery. At this time mineral lubricating oils were placed on the market. At first there was great opposition to these oils, but as the vegetable oils increased in price, tests were made which proved that a mixture of the fixed oils and mineral oils gave very good results. The percentage of mineral oil used in the mixture at the beginning was very small, but gradually increased until at the present time most compounded oils for marine-engine lubrication con-

tain about 80 to 90 per cent. of straight mineral oil and from 10 to 20 per cent. of fixed oil.

As stated above, olive oil was the favorite and most largely used at first, but when compounding or blending with mineral oils began, various other fixed oils were used. The principal ones now used in compounding are rapeseed, cottonseed, tallow and lard oils. The first two are usually blown, that is, air is blown through them, to increase their viscosity. The first named, rapeseed oil or rapeoil (usual blown), is used almost entirely in the compounded oils furnished the Navy, while lard and tallow are used largely in the cheaper compounded oils. The former contains less free acid and will not oxidize or gum as readily as lard or tallow oils.

In the past, most authorities considered it absolutely necessary to have a compounded oil for marine reciprocating-engine lubrication, as water was used in addition to the oil for cooling the bearings. All fixed oils form a heavy, creamy emulsion when mixed with water, while most well refined mineral oils will not form an *intimate* emulsion and, therefore, it was thought, would be easily washed out of the bearing. In addition, the fact that all fixed oils decrease very little in viscosity with increase of temperature as compared with mineral oils, was used as an argument for compounded oils. Also it was considered impossible to produce straight mineral oils with sufficient viscosity to maintain the oil film under heavy loads.

As long as the price of compounded oils remained just slightly above the price of mineral oils there was little thought of substituting the latter for the former, but with an increase of about 50 per cent. in the price of desirable fixed oils, it was considered important to determine by actual tests if a straight mineral oil could be used instead of a compounded oil.

The Engineering Experiment Station at Annapolis, Md., recommended to the Navy Department that the tests be made, and the tests were authorized.

A straight mineral oil was sent to the Experiment Station, and, after making the usual chemical and physical tests, the oil was tried out on one of the older battleships having reciprocat-

ing engines *without* forced lubrication. The compounded oil was used on the starboard engine and the straight mineral oil on the port engine, the conditions of lubrication being the same on both. The test was made on two runs of 590 and 216 knots each, at a speed of about 14 knots. The following results of the test were reported.

(*a*) "There were no warm bearings for either engine during the period of the test.

(*b*) "The *straight mineral* oil does not readily emulsify with water, in fact at no time did it form more than a 'skim milk' lather. This at first caused some nervousness among the oilers, but they soon realized that the bearings continued to run cool, and after awhile were in favor of the new oil, as it makes a cleaner looking engine; wearing surfaces, such as crosshead guides, were always clear and bright; not covered with a thick soapy lather.

(*c*) "The test oil, being lighter than the *compounded* oil, flowed more readily, and could be better and more economically applied at low speeds with few wicks.

(*d*) "The consumption of the two oils is about the same.

(*e*) "The present supply of *compounded* oil forms a very thick, creamy lather, which has to be washed out of crosshead guides and bearings very frequently. On the other hand, the *straight* mineral oil possesses practically no saponification quality, and at no time thickens or clogs up oil ways and bearing surfaces."

The following conclusions based on the above results were given by the Chief Engineer, Lieut. C. H. J. Keppler, U. S. Navy.

"As far as these limited tests of the new oil (straight mineral) are concerned it may be considered that:

(*a*) "The suitability of the *straight* mineral oil for service (naval) purposes on this type of engine is demonstrated.

(*b*) "The economy and efficiency as a lubricant of the two oils may be regarded as about equal. If any advantage, it is in favor of the *straight* mineral oil at moderate speeds.

(*c*) "While the *straight* mineral oil does not emulsify or form a thick lather, as long as a sufficient supply is maintained it serves as an efficient lubricant, and since it flows more readily the supply can be better regulated."

Although the above tests gave excellent results, it was thought necessary to make additional tests in order to study the action at higher speeds and under varying conditions. The Navy Department authorized tests on the U. S. Torpedo-boat *Bailey*, the test to be under the direction of the Engineering Experiment Station.

Two oils were selected for the test, one a *compounded* marine-engine oil supplied to the Navy for the present contract year, the other a straight mineral oil which was found to form an emulsion when mixed with water. This latter oil was not specially refined for the purpose, and was taken to represent the cheaper grades of straight-run engine oils.

These two oils were given the usual chemical and physical tests at the Engineering Experiment Station, the results of which are given in below in Tables I and II.

Table I.—Chemical Constants.

	Straight Mineral.	Compounded.
Loss of weight when heated 7 hrs. at 235 F.	1.43	1.53
Free acid, calculated as H_2SO_4, per cent.	0.00	0.00
Free alkali, calculated as NaOH, per cent.	0.00	0.00
Fixed oils, per cent.	0.00	21.00
Fixed oils, kind	....	Blown rape oil.
Qualitative Tests Showed:		
Free acid	None.	None.
Moisture	None.	None.
Matter insoluble in gasoline (86–88 Baumé)	Slight trace.	None.
Matter insoluble in ether-alcohol (8/6)	None.	None.
Free sulphur	None.	None.
Emulsifying constituents	Considerable.	Gives complete emulsion.
Gumming constituents	None.	None.
Sulphonated oils	Trace.	Trace.

Table II.—Physical Constants.

	Straight Mineral.	Compounded.
Specific gravity	0.932	0.940
Baumé gravity	20.215	18.936
Flash point, open cup, degrees F.	375	365
Flash point, closed cup, degrees F.	348	348
Fire point, open cup, degrees F.	412	414
Fire point, closed cup, degrees F.	382	400
Cold point, degrees F.	(+) 5	(+) 5
Viscosity, Saybolt, at 100 F., seconds	515	679
Viscosity, Saybolt, at 130 F., seconds	204	293
Viscosity, Saybolt, at 210 F., seconds	55	72

An emulsion test was made on the two oils, using the Experiment Station's emulsion machine, and stirring at 1,500 r.p.m. for 5 minutes, 40 c.c. of oil and 40 c.c. of salt water. The straight mineral oil gave a "skim milk" emulsion which settled out into oil and water after standing for about 40 minutes. The compounded oil gave a thick creamy emulsion which failed to settle out.

Figure 1 shows the viscosity curves of the two oils. It will be noted that although the difference in viscosity at engine-room temperature of 75 F. is over 400 seconds (Saybolt), the difference at bearing temperatures of 120 F. is only about 100 seconds. The above fact explains why the straight mineral oil flows more freely at all temperatures, and still is heavy enough to prevent its being squeezed out.

Before the practical test on board the *Bailey,* preliminary wick-feed tests were made using varying numbers of strands and a one-half-inch pipe. It was found that for this size oil pipe an increase in the number of strands up to *eight* gave increase flow of oil, but for ten strands and above the amount decreased. For this reason only eight strands were used on the tests on the *Bailey.*

Another preliminary test was made to show the relative rates of flow. The results of this test are shown graphically in figure 2. These curves show when using a half-inch pipe and eight strands, that the straight mineral oil flows about 60 per cent. faster than the compounded oil.

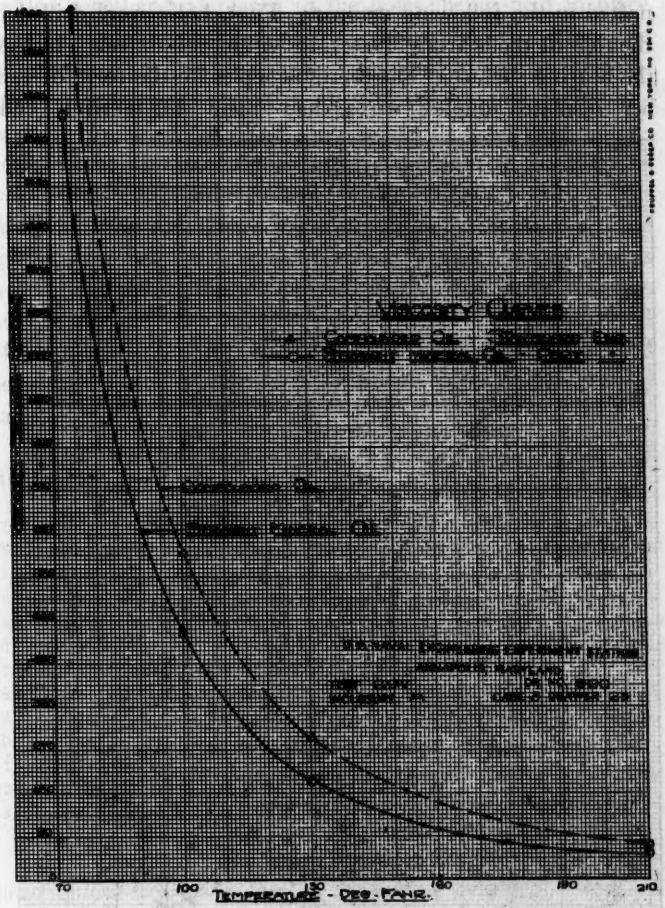


FIG. 1.

To obtain the desired information regarding the lubrication value of the two oils under practical conditions on the Torpedo-boat *Bailey*, an apparatus was rigged on the forward low-pressure crank and main bearings in order to secure the temperatures of these bearings under the different conditions. This apparatus consisted of iron-constantin thermo-couples con-

nected in the following manner: First, holes were drilled in the tops of the bearing brasses, to within one-sixteenth inch of the bearing surface. The thermo-couples, in the form of rubber insulated duplex wires, had their *hot-junction* ends

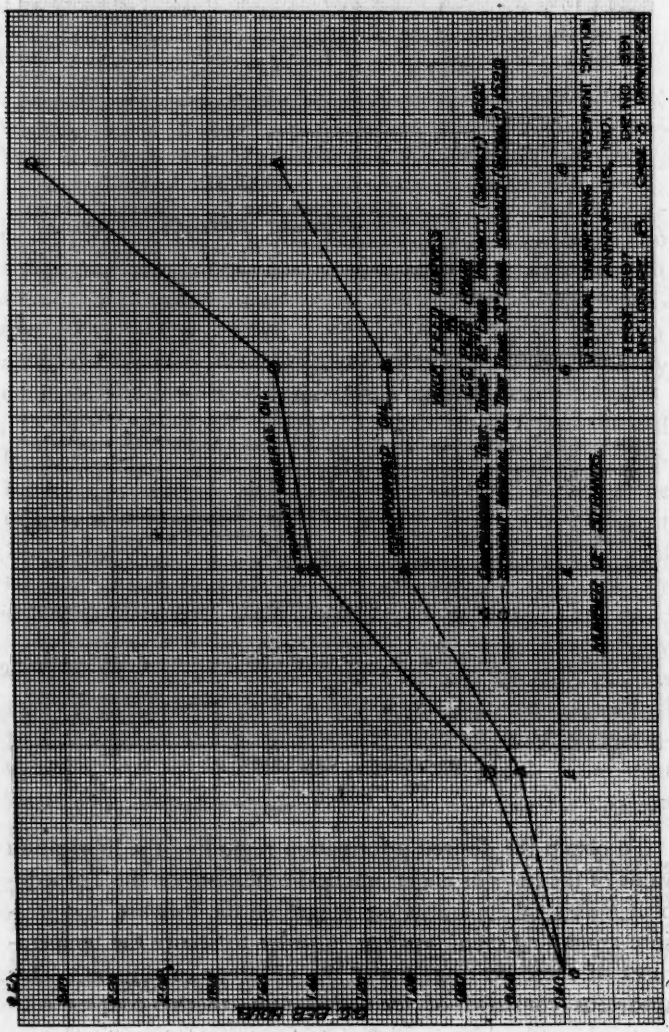


FIG. 2.

welded together and placed in the holes, being held in place by melted sulphur pounded into the holes around the wires. The other ends of the thermo-couples, commonly known as the cold-junction ends, were placed in an ice bath, so that their temperatures remained at 32 F. By means of a portable potentiometer the number of milli-volts generated by the thermo-couples was then read, and by referring these values to a calibration curve, previously made out, the temperature of the bearings in degrees F. was obtained. The thermo-couples were connected to the low-pressure crank and main bearings of each engine. Numerous runs were made covering a period of several months before satisfactory continuous readings could be obtained, due to the breaking of the wires leading to the rapidly moving crank bearings.

The following general data are given in connection with the test:

1. Number of engines..Two in same engine room, one starboard, one port.
2. Type of engine.....Vertical, four cylinders, triple-expansion.
3. Stroke .....18 inches.
4. Diameter of cylinders, inches.....20, 30½, 32, 32.
5. Diameter of crank, inches.....7.
6. Type of lubrication.....Wick feed.
7. Length of test, hours.....3½.
8. R.P.M. (average both engines).....182.
9. Boiler pressure, pounds gage.....220.
10. Vacuum, inches .....23.5.
11. Engine-room temperature, deg. F.....84.
12. Port crank bearing, drops of oil per minute.....17.8.
13. Starboard crank bearing, drops of oil per minute.....15.8.

The oil reservoirs for the crank pins, wrist pins and cross-head guides are made fast to the engine cylinders. The oil reservoirs for the main bearings are directly on top of the bearings. The crank pins are lubricated through oil pipes running through the connecting rods. The water service as installed permanently is of the older type, running through the crank from one end of the engine, but this was blanked off for the test, and a temporary pipe run with spray attachment, held in place above each crank pin. No water was used on the main bearings.

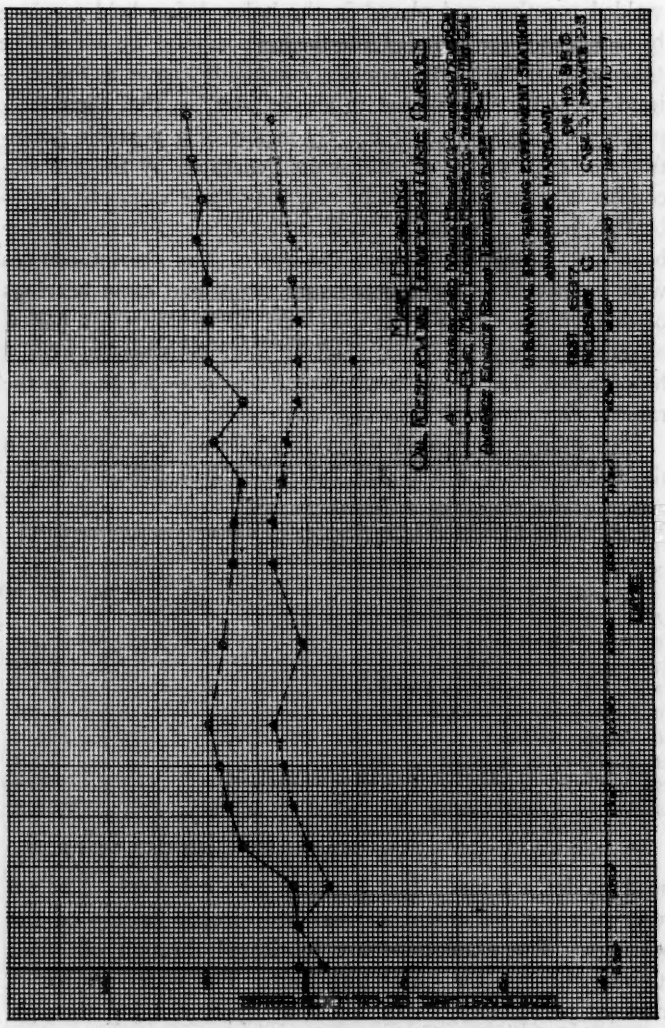


FIG. 3.

Figure 3 shows the temperatures of the *main-bearing oil reservoirs,* and figure 4 shows the temperatures of the *main bearings* for the period of the test. A comparison of the curves

shown on these two figures indicates that the temperature of the bearing is about 15 F. higher in both cases than the temperature of the reservoirs. This is to be expected, as the reser-

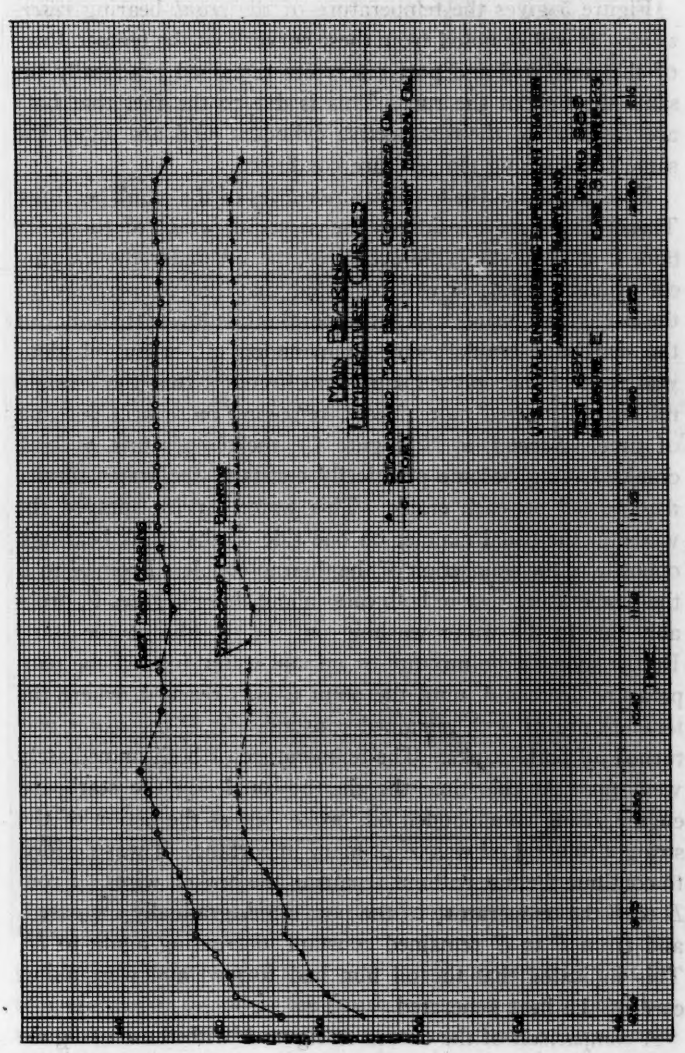


FIG. 4.

voirs are made fast to the bearing caps. The curves on figure 3 show that the temperatures remain practically constant for the main bearings after a short period of time.

Figure 5 gives the temperature of the *crank*-bearing *reservoirs.* It will be noted from these curves that the temperatures of the starboard and port reservoirs remained practically the same throughout the run. This is due to the fact that both are fastened to their respective cylinders, and are under the same surrounding temperature conditions.

Figure 6 shows the temperature curves of the crank bearings. These curves are considered the most important of those obtained, as they show the action of the oil under the varying conditions. *First,* it will be noted that the temperature of the port crank, using straight mineral oil, is lower than that of the starboard crank, using compounded oil, when oil alone was used. The only explanation offered is that the straight mineral oil flowed more freely, removing more heat per unit of time. *Second,* at *A* (figure 6), the temperature of both cranks having become practically constant, the water service *was turned on* each crank. This water service sprayed the water in the usual manner on one side of the crank bearings, opposite to the connection of the thermo-couple. From *A* to *B* the temperature rose in both cases about three degrees F, probably due to the water washing out the oil film at the start. But as soon as the mixture of oil and water formed the temperature decreased, until the point *C* was reached, where the temperature of both remained constant. At *D* the *water* was turned *off* both cranks. The temperature of both immediately went up. At this time the thermo-couple on the starboard engine, using compounded oil, broke, but as the action of the straight mineral oil was of primary importance, it was decided to continue the test without repairing the broken wires. From *D* to *F* the temperature of the port crank gradually increased, and from *F* to *G* remained constant, decreasing again to *H*. This indicated that the oil film had formed and was taking care of the heat generated.

A comparison of the curves on figure 5 with those of figure

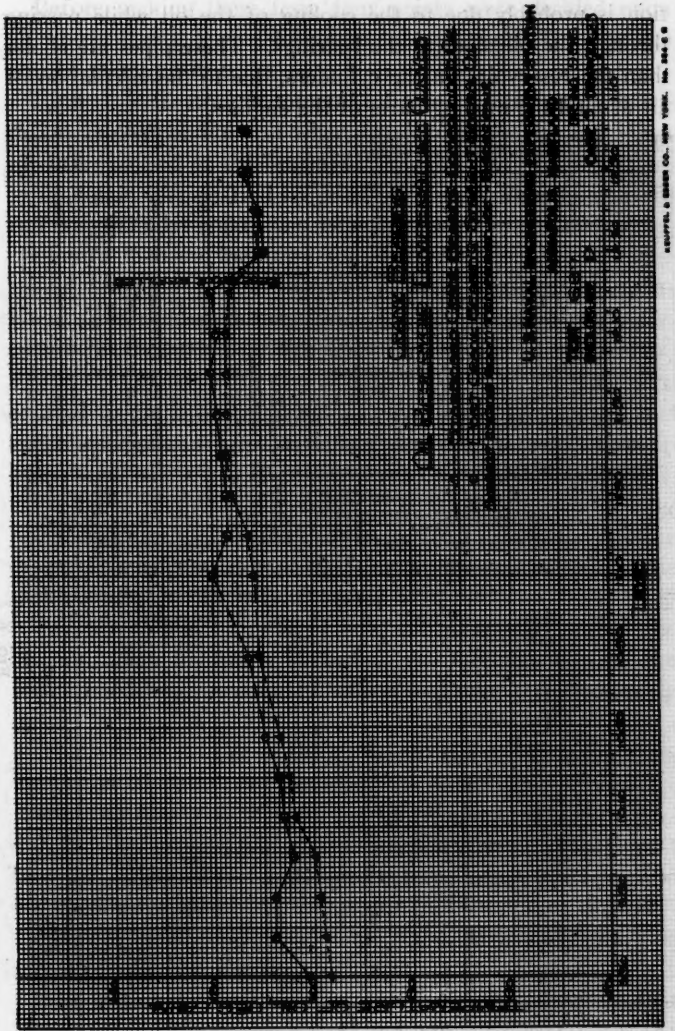


FIG. 5.

6 shows the existence of a peculiar condition; namely, that the temperature of the oil in the crank reservoirs was *higher* than the temperature of the bearings at certain times. This condi-

tion is probably due to the cooling of the oil while passing through the oil pipes to the bearings. These pipes are cooler than the reservoirs on the cylinders, due to the circulation of air around the moving parts of the engines.

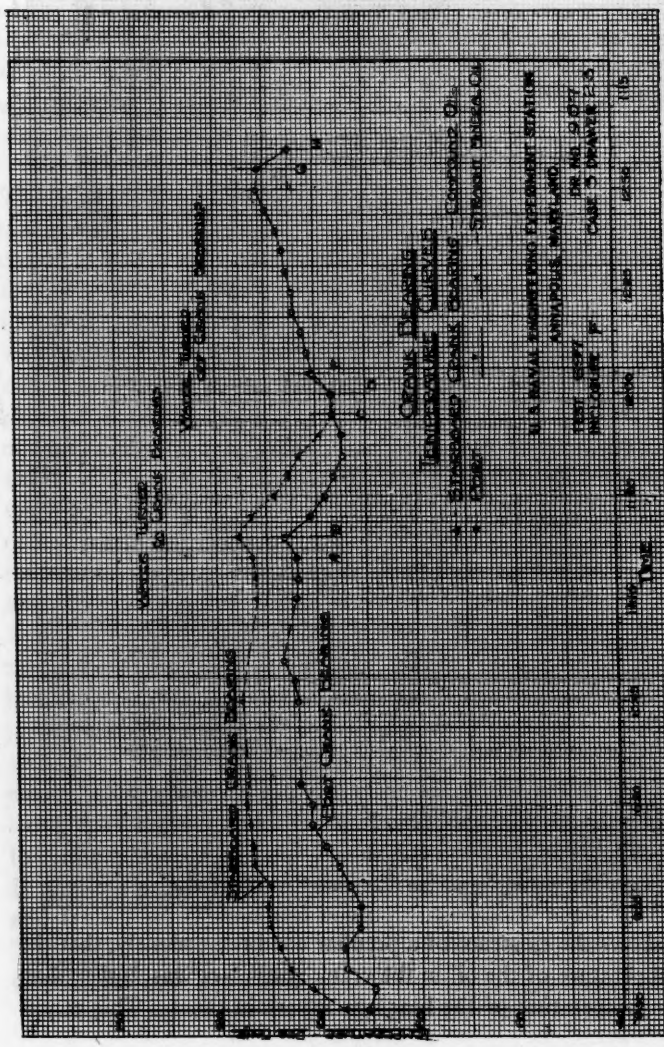


FIG. 6.

Before making these tests four points were decided upon on which it was desired to obtain information. These points will be taken up in order.

First: "Will straight mineral oil give as satisfactory lubrication as compounded oil when fed through wicks and when *no* water is used on the bearings?"

To obtain information on this first point, both engines were run under similar conditions using *wick* feed *only*. The curves on figure 6 from the beginning to point *A* show that the straight mineral oil lubricated more efficiently; that is, maintained lower temperatures than the compounded oil. The curves for the main bearings figure 4 gave the opposite results, which are explained by the fact that the oil reservoirs were secured to the bearings; also both main-bearing temperatures remained constant after a short time. From the above it is considered reasonable to believe that the two oils have practically the same lubrication value; that is, will prevent seizing of the bearings, reduce the friction in direct proportion to their viscosity, and prevent rapid heating of the bearing above the melting point of the bearing metal.

Second: "Will straight mineral oil be washed out of the bearing *when the water is turned on,* thereby allowing the journal and brass to come in contact and wipe out the bearing metal?"

The section *A* to *D* of the curves on figure 6 prove conclusively that turning water *on* the bearing will not destroy the lubricating film, also that the bearing temperatures will be reduced. The curves from *A* to *B* on this figure indicate that at first the oil is washed out, but that the mixture of oil and water soon reduces the temperature.

Third: "Will straight mineral oil form a continuous oil film after the water is shut off?"

The section *D* to *H* of the curves on figure 6 shows that on turning *off* the water, the oil film will form very soon, and as soon as the normal bearing temperature has been reached the temperature will remain nearly constant.

From the results obtained on the tests enumerated above, the following conclusions were reached:

That at high speeds (180 r.p.m.) straight mineral oil will give as efficient lubrication as compounded oils.

That water running on the bearings will not break the oil film when straight mineral oil is used, and will cool the bearing, the mixture giving efficient lubrication.

That when the water is shut off the oil film will form, using either straight mineral oil or compounded oil.

That any straight mineral oil which will emulsify when mixed with salt water will give satisfactory results when used for the lubrication of marine reciprocating engines *not* having forced lubrication.

It is known, of course, that some merchant ships now use straight mineral oil and obtain satisfactory results, and it is thought that it will only be a question of time when all engines without forced lubrication will use it, having due regard for any peculiar or exceptional conditions, which may necessitate the use of compounded oils.

RATIONALITY IN PHYSICAL AND METALLOGRAPHIC TESTING.

By K. F. Smith, Lieutenant (J. G.), U. S. Navy, Member.

There is today no branch of mechanical engineering more important than that which has to do with the physical testing of the materials of construction. The rapid growth in demands for reduction in weights with increased strength has been one of the hardest conditions the manufacturer has been obliged to meet. In meeting this condition he has had to rely on antiquated methods of testing which are, to say the least, of doubtful value in certain cases.

Happily, a method which is as yet in its infancy, is springing up as a guide to the mechanical engineer in his work; a method which shows him quite clearly what conditions he must meet, and what actually happens to his material under various conditions of use. This is the science of metallography. Unfortunately, the present-day engineer knows nothing about this new "highbrow fad," and in many cases is unwilling to learn. On the other hand, the metallographer is all too often a pure scientist who considers an analysis of stresses beneath his dignity.

Both types will in time disappear, and we shall have designers and testing engineers who use metallographic methods as freely as they do physical ones, and in some cases to better advantage. We will have designers who, in drawing up specifications, will call for low-carbon steel when low-carbon is satisfactory, and nickel-chrome when that is necessary; but who will not, just because the particular part is subject to shock, specify a ductile material when a high-carbon steel is necessary.

A case of this kind has recently been called to my attention. A prominent firm producing rock drills had, as is customary, been specifying wrought-iron or low-carbon steels for a certain part of their drill which was subject to intense shocks. This part gave continual trouble and required frequent renewals. One of their engineers happened to be in the field with a drill, when this part broke, and he found that he was without a "spare." It was an emergency job, and repairs had to be made. As a last resort a high-carbon steel part was turned out and installed, in the hope that it might last for the job. It did. Today that firm is using high-carbon steel for all their shock parts.

How often, too, do we hear an engineer say that a defective part has recrystallized and failure is due to this cause; and how hard it is to convince him that a metal *cannot recrystallize due to vibration.* It may, it is true, fail through fatigue, but that is a very different thing, and a fatigue failure can usually be discovered under the microscope.

We in this country are behind the times in regard to materials of construction, and this is due to two main causes: First, we do very little research ourselves, but rely too much on the results of others. Second, the manufacturer seems averse in a majority of cases to allow a complete examination of the defective material of his firm. Without a close study of breakdowns and defective material, and the circumstances which have led up to the defect, and without free discussion of cause and effect in metallurgical operations, we can never hope to arrive at any degree of proficiency in this line. It is true that today a great many firms had rather waste thousands of dollars a year in defective material than spend hundreds in scientific research.

We are further behind in the testing field than perhaps in any other. I believe the manufacturer can produce practically any material we want if we only know ourselves—if we would only draw up our specifications to cover *what we want!* I have recently seen a specification for steel drawn up which it would be physically impossible to meet by the method of manufacture specified; but had the purchaser left it to the discretion

of the manufacturer as to how he made that particular steel, he would have received a material which would have been satisfactory in all respects, and met all the physical requirements necessary.

It seems to me that, before drawing up a specification, we should first study carefully just what use is to be made of the material; what type of stress the part is to be subjected to; then specify, with a full knowledge of the structure and characteristics of metals in general, the metal to be used, its composition and treatment, and, finally, the methods of testing the finished product.

If a part is to be used in practice as a member subject to alternating stress, why not make some test which will show how that particular part will behave under this stress, confirming this by other tests, if any are known? If it is to be subject to alternating impact, why not test for alternating impact? From what little we know now of the behavior of metals under various stresses, we can say that a tension test does not necessarily show how the metal will behave under alternating stress, and yet it is usually the test prescribed.

To understand fully why different tests are necessary we must study, first, the structure of the metal itself, as revealed by the microscope, and the changes in that structure due to various types of stresses.

As we all know, metals are crystalline, that is, made up of an infinite number of "grains" cemented together by what Beilby calls an amorphous cement. The size of the grain depends upon various factors: the casting temperature; the rate of cooling; thermal treatment; the amount of work done on the material, either in the rolls, or in the press or forge. These grains are, in turn, made up of millions of crystal units, each one oriented in the same direction within the grain, but the orientation varying from grain to grain. Why they should have the same orientation within a grain; why the crystals of most metals should belong to the cubical or regular system; or why they should be crystalline at all is, of course, unknown. The most

plausible explanation seems to be that in the liquid state infinitesimal particles, each with a certain electric charge, move around with great rapidity. We can imagine them to be minute magnets. As the temperature falls the velocity of movement decreases, and three or four meet and behave as magnets would, that is, assume the same orientation. This action is going on at various points throughout the liquid, and we have a series of nucleii. As the temperature falls still further, movement becomes more and more restricted, and a sort of magnetic field springs up around each nucleus, and other small particles are attracted to these centers, and a grain gradually builds up. There being thousands of centers of crystallization at once in a more or less viscous liquid, and each having a slight magnetic field around it, the tiny crystals assume a direction parallel to the field direction at that point, and we have them oriented in constant directions in each field. New particles coming in contact with the nucleii are affected by the field of that nucleus, and assume the orientation of the incipient grain. The grains continue to grow in all directions until they meet the one next adjacent, and then stop and grow in the free direction. As the grains are built up of tiny cubes or octahedra it is seen that, owing to the difference in orientation, there would be a stepped space left vacant. However, there is a little liquid left there, and, due to the close proximity of two different grains and the different magnetic influences, the infinitesimal particles are twisted in all directions, and, as the temperature is still falling are finally caught before they decide which way to orient, and go solid as an amorphous cement. This cement may be influenced slightly by intergranular pressure. That this amorphous cement actually exists is confirmed in part by the fact that a break does not follow grain boundaries, but crosses them, showing that the boundary is actually stronger than the grain itself; and further by the fact that chemical reagents act differently with respect to the boundary than to the grain.

We must now examine one or two properties of crystals in order to understand what follows.

It has long been known that the properties of crystals vary in different directions. They are harder in one direction than in another; the solubility in chemicals or in water is different. If a sphere be turned out of rock salt and suspended in still water it will be found that the sphere turns into an ellipsoid, due to this difference in solubility; and it is partly due to this property that we are able to etch the surface of our polished metal for metallographic work, the reagent acting on one grain more than another, depending upon the direction in which the grain is cut, and thus forming a stepped surface. Electrolytic action between grains and the rejection of impurities to the boundry also aids.

It is noticed that the brittleness varies, and that if we try to break a crystal the break will occur along planes which are usually parallel to the crystallographic axes or a crystal face. These planes of weakness—called cleavage planes—are of great importance, as it is along these planes that incipient fracture in tension occurs.

We are now ready to see how straining a metal affects its structure.

Below the elastic limit, and that is really the point where the applied stress equals the intermolecular cohesion of the crystal on the cleavage planes, the crystals seem to be perfectly elastic and the metal, if only strained but once, seems to be perfectly elastic, and we have a strain proportional to the stress producing it. It is readily seen that in a conglomeration of crystals oriented in various directions, some crystals are so located that their plane of weakness coincides with the direction of force application, and when the stress exceeds the elastic limit of the grain an incipient slip occurs across this plane of weakness. Naturally this throws an added stress on adjacent crystals, and if their orientation is such that they are in a poor position relative to this intensified stress the slip is transferred, and finally fracture occurs.

We can imagine the grain to be like a pack of cards—the direction normal to the card surface corresponding to the direc-

tion of maximum crystalline stiffness, and the card surfaces being the direction of the slip planes. Apply a stress normal to the surface and nothing occurs but a compression of the pack, which disappears on removing the stress. Now, incline the line of application and the entire deck slides out; that is, our grain has given way under a lighter stress, applied in a direction having a resultant in the direction of one of the intercrystalline weakness planes. Our individual card has not changed in shape but has simply slid along. Figures 1 and 2 represent the microscopic appearance of a metal before and after strain beyond the elastic limit.

The steps apparent on the surface are called slip bands. That this is what has occurred in metals is shown by the appearance of bands on annealing strained metal. As a usual thing these bands are straighter and more regular in lead, tin, silver and gold than in iron or mild steel, where they are often curved, irregular and branched.

We can now see how metals fail under tension, but one fact remains to be explained. Why is it that a metal after over-

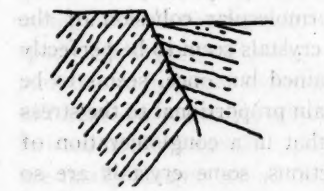


Fig. 1.

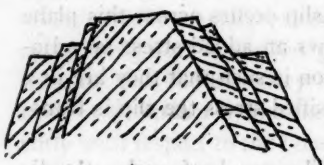


Fig. 2.

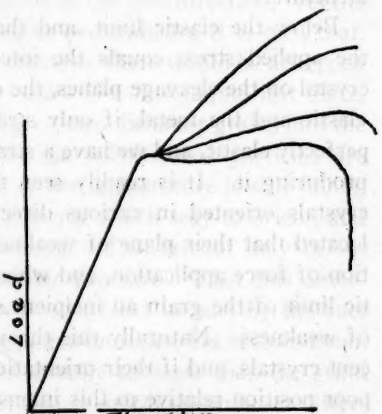


Fig. 3.
Variation in tail of curve with changes in $\frac{L}{D}$

strain beyond the elastic limit will show a higher elastic limit on another application of load? If what I have said about slipping is true, certainly something else must have occurred to raise this limit, as the grain cannot have turned in the meantime to present a new face.

Beilby says that when a slip occurs, due to a molecular disarrangement, a viscous amorphous variety of the metal is formed along the slip plane, which at first acts as a lubricant, facilitating the slipping. This amorphous layer gradually sets, and the slip on this plane stops, and a new band must develop. Now all evidence points to the fact that this amorphous layer is actually stronger than the crystal itself, and acts as a stiffener. On straining a second time, a greater stress would be necessary before further slip occurred along new planes. As an evidence that this occurs, the plasticity of a metal at its yield point is often cited. The formation of this amorphous layer and the slip takes a certain amount of time, and in the case of shock the application of the load is so sharp that slip does not have time to occur, and the grain breaks along the cleavage planes.

In alternating stress the first action is similar to a direct tension with the formation of the amorphous layers. If now the stress is maintained constant, due to the setting of the amorphous cement, no further slip occurs, but if the stress is reversed while the amorphous layer is still mobile the layer thickens, and after a time is actually forced out, and an incipient crack develops, throwing the stress onto adjacent grains which then fracture in turn.

I may say in passing that there is considerable opposition to this "Amorphous Theory," but it is mainly a question of dimension rather than of fact.

What this amorphous cement is is a matter of conjecture, but the consensus of opinion seems to be that it is really an undercooled liquid, and much like glass in this respect. This is confirmed partially by the fact that at high temperature the resistance is a function of the rate of strain, and thus obeys the laws of viscous liquids; and again by the fact that, if this ce-

ment is an under-cooled liquid, it should show some viscosity slightly below its melting point. Rosenhain has produced a perfectly brittle fracture in lead, tin, and even gold, at from 3 to 10 degrees below their melting points.

We have now seen what occurs in a metal in use, and can readily see that a tension or a bending test of a material to be used in a position where it will be liable to alternating stress, gives little information as to what will occur in service. This, for example, would be true of shafting to a slight extent; to connecting rods and a hundred-and-one other machine parts in everyday use.

As the question to be decided practically is: what is the limiting stress I can safely use in design? We *must* find the stress limit under these various types of load. The ultimate strength and yield point, while in themselves interesting to study, are not useful as design factors. What we want is the elastic limit.

How often engineers in their computations say, when given the ultimate strength of a metal: Let us assume that the elastic limit is half the ultimate strength. This is, however, a dangerous practice, as the elastic limit falls as low as .26 of the ultimate in extreme cases. Frequently, in giving values for the elastic limit, what is actually given is the yield point, which is always above the elastic limit when the yield point is shown at all.

In testing in tension or compression, we should then specify, not only the form and dimensions of the test piece and rate of load application, but also the method of determination of the various points. The effect of the dimensions of the test specimen are greatest on the ductility figure; and we may say that elongation on ductility figure is worthless unless we know the dimensions of the test specimen. It has been found experimentally that if the ratio of $\frac{L}{D}$ exceeds 3, the tail of the load curve only is affected by the length over which elongation is measured (Fig. 3); but if under 3, the entire curve is affected.

The reduction in area is the more reliable method of judging the ductility characteristics of the metal in any case.

The apparatus for testing in tension, compression or torsion is satisfactory, and our methods, when correctly applied, will give the factors necessary for correct design and these values—with the factors of safety selected to cover defective workmanship and other possible defects or uncertainties of stress distribution,—may be used directly. We must, however, use discretion in estimating what our maximum stresses are to be. For example, it is next to impossible to apply a concentric load, and the investigations of Baraclough have shown that the stress distribution under eccentric loading due to the combination of direct tension and bending, throws the neutral axis to the tension side. In compression due to non-parallel ends, and the resulting shear, the actual stress is increased by 20 per cent. or more. The effect of discontinuities in a tension or compression member is also important, and without going into the theories, we may say that if we have a cylindrical hole in these members, the stress is tripled at the boundary; and if the hole be elliptical, with the shorter axis in the load line, the stress is increased by an amount equal to twice the ratio of the axes. For this reason it is extremely bad practice to tap a hole in the skin of shafting for any purpose.

In the first part of this article I stated that a high-carbon steel is preferable to a wrought-iron for shock parts, providing the stress intensity does not exceed the elastic limit. This will, however, not be accepted without a proof. Metallography shows us that, under a shock, the metal actually extends to the point corresponding to the elastic limit, and that up to this point the deformation has been proportional to the load, and equal to that produced by a steady load twice as great. In this case, however, slip does not occur, due to the speed of application, and the metal breaks slightly above the elastic limit. If we call the work done in stretching the material up to the elastic limit: *i. e.*, the area under the load curve to that point, the "Elastic Resilience," we see that this is a measure of the shock-absorbing power.

Let P = load in lbs. at E.L.

p = load in lbs. per sq. in. at E.L.

A = area of section.

L = length of bar.

λ = stretch per unit length.

E = Young's Modulus.

R = Elastic Resilience.

$$\text{Total stretch} = L\lambda = L\frac{p}{E} = L\frac{P}{A} \div E.$$

$$\text{Area OEB} = \tfrac{1}{2} \text{ load} \times \text{elongation} = \tfrac{1}{2}pA \times L\frac{p}{E}.$$

$$R = \tfrac{1}{2}\frac{p^2}{E} \times (\text{volume of piece}).$$

We thus see that the resistance to shock is a function of the square of the elastic limit, and as E 30×10^6 for steel and 28×10^6 for wrought-iron, a difference of only 7 per cent., while p increases rapidly, it is easily seen that the high-carbon steel is best for shock if free from other stress actions. *

If we call $\frac{\frac{1}{2}\ p^2}{E}$ the Modulus of Resilience in inch-pounds per cubic inch, the shock-absorbing power is given at once by multiplying this modulus by the volume in cubic inches. By this method the following table is obtained.

* The need of considerable ductility in gas containers to allow considerable distortion before fracture is recognized, but this ductility is necessary to produce a tearing failure in place of shattering.

The Resilience of Open-Hearth Steels.

(Courtesy of Prof. Walter Rautenstrauch.)

Per cent. Carbon.	Yield Point.	Mod. of Resilience in pds. per cu. in.
.10	32,000	17.
.20	38,500	24.7
.30	45,500	34.5
.40	52,000	45.0
.50	58,500	57.0
.60	65,000	70.4
.70	71,000	84.0
.80	76,000	96.3
.90	82,000	112.
Vanadium Steel	170,000	481.
do.	195,000	634.
do.	225,000	844.

In compression, torsion or bending, this Resilience factor is the mark of the shock-resisting power, Young's modulus for shear, however, being used in torsion, and in each we find the shock-absorbing power to be a function of the elastic limit.

BENDING.

There is no test used more today, with the exception of the tests mentioned above, than the bending test, and at the same time there is none which gives more valueless results, as usually made, than this very test. When the test is used as an auxiliary, as, for example, in bending a boiler tube through 180 degrees on a specified radius, without the development of cracks, we are taking cognizance of the future treatment of this tube in practice, and the test is valuable in that it shows that the material will stand that treatment. It is not, however, quantitative.

Suppose we have a member which, in use, will be subject to pure bending. It is manifest that we must know how this material will behave in use. We could compute the stresses en-

countered and take the elastic limit from our tensile specimen, but the question is: Does this represent the elastic limit in bending? The answer is: It certainly does not in certain cases. If materials of engineering were homogeneous, in all probability the results of a tensile test would be satisfactory; but a microscopic examination of longitudinal and transverse sections shows that the structure is different; we may have a segregated center in rods or shapes, and certainly in the case of rolled material, the slag, manganese sulphide, and in the case of cold work even the grain itself, is drawn out in a logitudinal direction, and in all probability the physical characteristics are different in these two directions.

In all tests so far considered there are two main objects: first, the determination of the physical constants; and second, the detection of faulty material. In the bending test as usually applied there is little chance of detecting faults, as will be seen.

The usual method of making this test is to place a bar of standard dimensions upon knife edges at a known distance apart, and apply a central load, measuring the resulting deflection, and from that computing the maximum fiber stress. (Fig. 4.) Let us examine the Bending Moment Diagram in this case, and we find the triangular diagram A B C. The deflection is equal to $\frac{PX}{48EL}(3L^2 - 4X^2)$, where X is the distance from supports. This is seen to be a maximum when $X = \frac{L}{2}$. By measuring the deflection we may compute E, and hence the fiber stress at failure. Now, unless the defective part happened to be directly under the applied load, there would be no indication of its presence, as the fiber stress reaches the maximum value at one point only. Let us use a method shown in Unwin, and see what happens. This method is shown diagrammatically in Fig. 5. Certainly there is no practical difficulty involved in the test. Examine this Bending Moment Diagram, Fig. 6, we find that between B and F the Bending Moment is constant and equal to Px. Examining further, we find that,

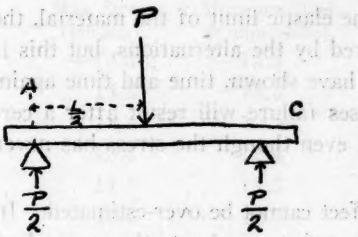


Fig. 4.
The present method

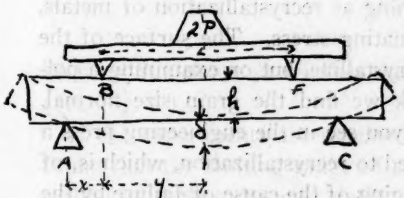


Fig. 5.
The new method

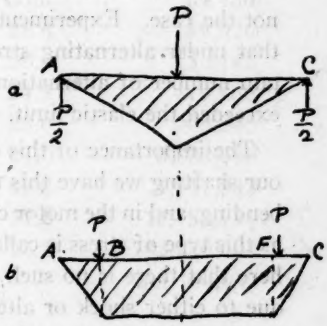


Fig. 6.
Bending Moment Diagrams
(a) Present method
(b) New method

under this loading, B F will bend in a circle, and the fiber stress at any point between is $\frac{h}{y^2 + 2}$. If there is a defective spot between the points B and F, the localization of stress due to the defect will cause failure at that point, and a microscopic examination will usually locate the cause of the failure.

In this case we have accomplished both objects of the test, the determination of the physical constants, and an estimate of the homogeneity of the material.

ALTERNATING STRESS AND FATIGUE IN SHAFTING.

We have seen the effect of alternating stress on the microstructure of metal, and it is certainly plausible to expect that a tension test would not show the behavior of a part subjected to this type of stress. We might expect that, if we did not

allow the stress to exceed the elastic limit of the material, the material would not be injured by the alternations, but this is not the case. Experiments have shown, time and time again, that under alternating stresses failure will result after a certain number of alternations, even though the stress has never exceeded the elastic limit.

The importance of this effect cannot be over-estimated. In our shafting we have this type of stress, due to the reversal of bending, and in the motor car industry breaks due to " fatigue," as this type of stress is called, are frequent. It should be noted here that there is no such thing as recrystallization of metals, due to either shock or alternating stress. The surface of the break often looks coarsely crystalline, but on examining a polished specimen at the break we find the grain size normal. Hardly a week goes by but you see in the engineering press a record for a failure, attributed to recrystallization, which is, of course, impossible. The judging of the cause of failure by the appearance of the break, unless confirmed by other proof, is extremely dangerous.

In reference to the behavior of materials under repeated stresses, the experiments of Wöhler are classic, and have been repeated by numerous other workers with similar result (See " Engineering," Vol. XI, 1871. Also " Brit. Assoc. Report," 1887, p. 424.)

Summarized the results are as follows:

1. The resistance under a fluctuating load depends upon the algebraic difference between maximum and minimum stress, and not on the maximum.

2. Reversed loads will cause failure at a point well below the ordinary elastic limit.

In a steel showing an elongation of 20 per cent. in 2 inches and a tensile strength of 23 tons per inch, a test was made by weighting a round section and revolving it in such a way as to give the stresses shown in the following table (Moreley, Strength of Materials):

Max. Stress (tension).	Minimum Stress (compression).	Range, Tons.	Repetitions to Fracture.
+15.3	—15.3	30.6	54,430
14.3	14.3	28.6	99,000
13.4	13.4	26.8	183,145
12.4	12.4	24.8	479,490
11.5	11.5	23.0	909,840
10.5	10.5	21.0	3,632,588
9.6	9.6	19.2	4,917,992
8.6	8.6	17.2	19,186,791
7.6	7.6	15.2	132,250,000
			not broken.

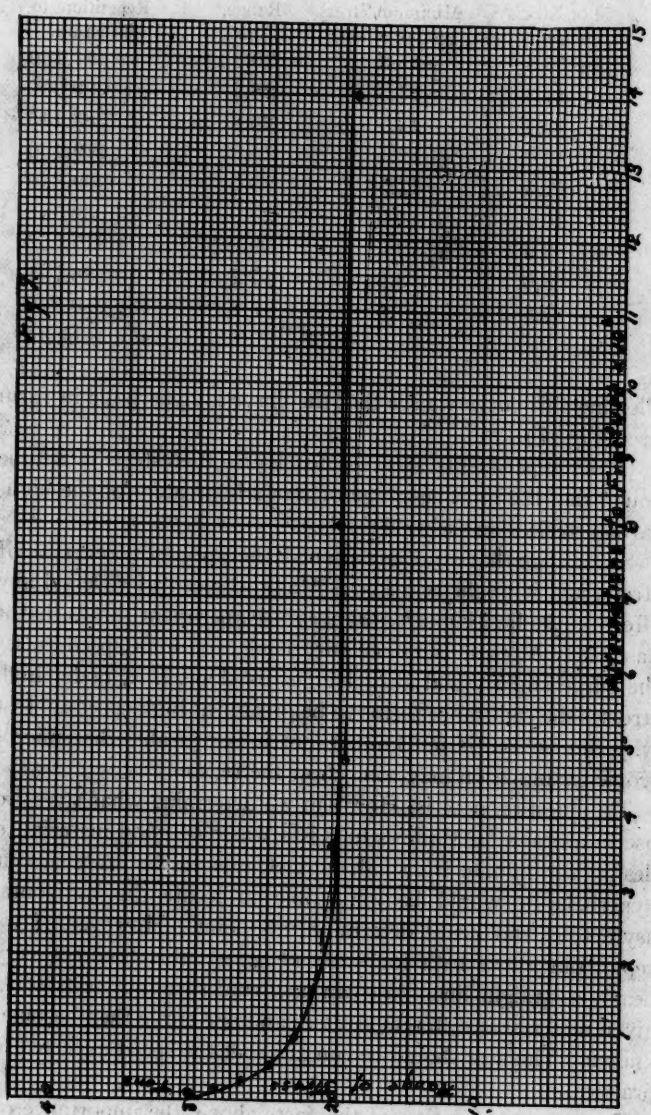
Plotting this curve we see that the curve is hyperbolic in form (see Fig. 7).

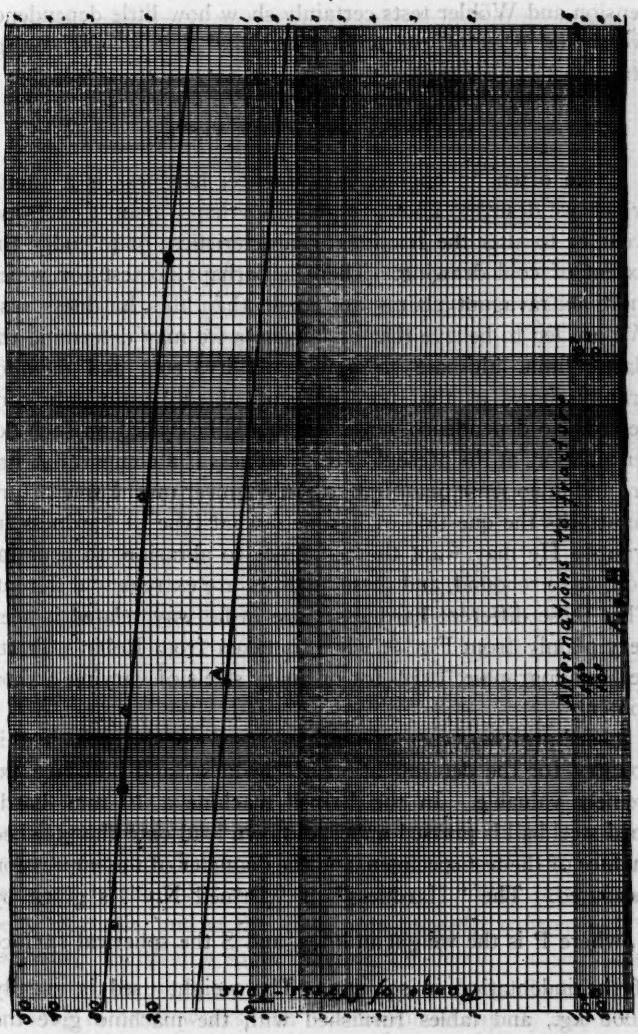
If the experiments are repeated with other steels it will be found that the limiting range of stress is greater for the high-carbon steels than the mild steels.

So far we have not mentioned the speed of reversal of stress, and it might be asked: Has the speed of reversal any effect on the limiting stress ranges? Experiment shows that it has, but the result is rather surprising, for high speed raises the curve slightly and also changes its slope, *i. e.*, under high stresses the endurance seems to be greater at high speeds. Just why this is true is not known, but it is probable that microscopic examination may soon show the reason.

The question now is, How can we determine what stress to use in a design? Manifestly it is a waste of good material to design for infinite life, and, in fact, due to the scale of our hyperbola, it would be impossible to locate the horizontal asymptote, and hence determine this point. However, if we replot our curve to a log scale (Fig. 8), this becomes very nearly a straight line. We can now run, say, four tests for a given material at different stresses and our curve then becomes known. This test should be run at about the speed expected in practice.

From a consideration of the facts above, the importance of





knowing the physical characteristics of a material to be used for shafting subject to excessive bending action, *i. e.*, long, unsupported shafts, is readily seen, and a comparison of static

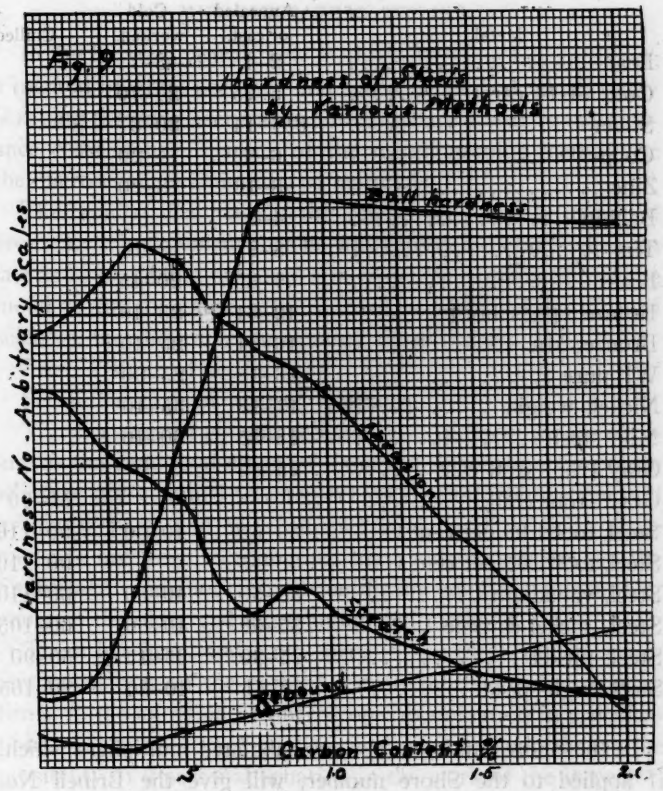
tension and Wöhler tests certainly show how little dependence we should place on the tension test, and that an alternating-stress test is necessary.

HARDNESS AND OTHER TESTS.

Quite often, in engineering practice, hardness is a factor which we desire to know, to guard, perhaps, against excessive wear or to alter the frictional characteristics; or to judge the extent of cold drawing. There is perhaps no test for which more extravagant claims are made than this hardness test. We are led to believe that by hardness we may determine the elastic limit, ultimate strength, ductility, etc., and metallography shows us that they are undoubtedly intimately connected. No dependence, however, should be placed on such determination, at the present date. Hardness is a comparative figure and in no sense quantitative. An engineer must know before he attacks the question of hardness what type of hardness he wants. He may want the resistance to abrasion, to penetration, or to rebound, and these qualities are extremely different. That these different hardnesses are in no way related is seen very plainly from Fig. 9, which is taken from Robin's Traité de Metallographie, 1912, for ordinary steels of varying carbon content.

There are two types of machines used considerably in this country for the determination of hardness, and both are extremely valuable. In the Brinell machine a 10 mm. hardened-steel ball is pressed under a known load onto the polished surface of the test specimen, and the diameter of the impression measured. The "Hardness" or "Brinell No." is given by $\frac{P}{S}$, where P is the weight in kilos, and S the spherical surface of the indentation in sq. mm. The weight used is usually 3,000 kg., and tables furnished with the machine give the "Brinell No." corresponding to the diameter of the depression.

In the other type, known as the Shore scleroscope, a dia-



mond-pointed hammer drops onto the surface of the specimen from a constant height. The rebound measured on a scale attached is the "Scleroscope Hardness." This instrument, being portable, and at the same time ruggedly constructed, is extremely useful in a shop. With it the depth of case in case hardening, the degree of temper in tool making, the effect of drawing in brass works, can be determined rapidly and accurately. But it must be remembered that the results are relative. With the Shore instrument the hardness numbers for various materials is given below. These results are taken from the Shore catalogue.

Metal.	Annealed or cast.	Cold worked.	Chilled.
Lead	2–4	3–7	
Gold 24-14 carat	5–25	24–70	
Silver	6½–14	20–37	
Copper	6–8	14–20	
Zinc	8–10	18–20	
Babbitt	4–9		
Tin	8–9	12–14	
Brass	7–35	20–45	
Bronze (phosphor)	12–21	25–40	
Bronze, Mn.	16–21	25–40	
WI, pure	16–18	25–30	
Nickel, weight	17–19	35–40	
Mild steel	18–25	30–40	
Gray Iron, sand cast	25–45		
Gray Iron, chilled			50–90
Steel, tool 1.00 per cent C	30–55	40–50	90–110
Steel, tool 1.65 per cent C	38–45		90–110
Steel, vanadium	30–50	40–60	50–110
Steel, nickel-chrome	35–50	40–60	60–105
Steel, nickel	25–30	35–45	50–90
Steel, high-speed	30–45	40–60	70–105

In the Shore catalogue also are tabulated multipliers which, if applied to the Shore number, will give the Brinell No. The values are as follows:

	Multiply Shore No. by
Tool steel, hardened and tempered	6.6
Drill rod	6.3
Brass, medium hard, drawn	5.6
Tool steel (annealed)	5.5
Mild steel (hot rolled)	5.25
Brass, drawn, annealed	5.0
Cast iron	4.6
Mild steel, cold rolled	4.6

These are important for comparison of current literature.

METALLOGRAPHIC TESTING.

In the last few years a new method of testing has appeared; a method of great accuracy and speed; a method which today is saving the purchaser thousands of dollars a year in maintenance costs alone. I speak of the metallographic method, or the microscopic examination of materials.

The rapid strides made in this science may well be imagined from Sorby's statement that he was considered crazy when, as late as 1890, he first proposed a microscopic examination of the metal to fix the cause of a railroad accident. Like all new sciences, metallography has passed three stages:

First, a stage of scorn.

Second, a stage of amused tolerance.

Third, a stage of general acceptance. And now we are rushing to the semi-final stage of fanatical application by people who are not qualified to interpret its results. This stage will gradually pass, and with further study and publicity these methods are now receiving we may soon expect to see the final stage: metallography scientifically applied, supplanting many of the older and more antiquated methods.

There has been, and I am sorry to say there still is, a violent opposition on the part of certain shortsighted producing firms, to the adoption of microscopic examination in specifications. They use every possible argument to suppress it; the possibility of a personal equation in the work of the inspector, the inexactness of the science as it exists today, and many others, equally foolish. I think, however, that it is now well established that metallography is one of the most exact test methods known. Not long ago I was talking to the metallurgical engineer of a steel plant, and asked him what metallographic methods he used. His answer was instructive. "The firm is opposed to metallographic examination." On asking why, he replied that it caused too big a percentage of "seconds." I then asked what they did when they knew the purchaser used this method, and the reply was equally instructive. "Oh, of course we use it then."

The layman seems to think that a metallographer can eliminate entirely the mechanical tests necessary for acceptance of a material; that he can, by looking at *one* specimen from an important forging, tell just what the metal will stand in practice; what its physical properties are; that he can tell how the metal was made; what treatments it has had and what further treatments are necessary. Some metallographers make this error themselves; all too frequently we see examples in the engineering press of "snap judgment" on the part of the metallographer; statements that are not justified by the evidence in hand. The successful metallographer must be thoroughly familiar with the theory of metal structures; must understand some physical chemistry; must be a metallurgist, and know at least what stress means. He must, in fact, be both a theoretical and a practical engineer.

No metallographer claims that his science is the acme of testing methods, but rather that *it is an important auxiliary method,* and one which, when properly applied, will safeguard the interests of the purchaser to a greater degree than any other.

Osmund very aptly compares metallography with medicine, and divides the science into three groups:

Histologic-anatomical.
Biological.
Pathological.

The first group deals with the crystalline structures, their dimensions and characteristics; the second, the forms, dimensions and relations of the different constituents in relation to the thermal or mechanical treatment in the course of manufacture; and the third is a study of faulty treatment of impurities which cause certain diseases, and the relation of cause and effect in faulty material.

The practical metallographer must deal with all three. Before he can give an exact opinion of a cause of failure he must know the entire history of the material he is examining. Without this he is working in the dark. The importance of knowing the complete history of the material cannot be over-

emphasized. True, a good many of the points in the manufacture, and subsequent thermal and mechanical treatment, are visible at once under the microscope, but without an exact knowledge of the metallurgical methods the metallographer is working under the same handicap as a specialist in human diseases who does not know the past history of his case.

A great point, and one which has not received sufficient attention as yet is, from where should metallographic samples be taken? The adage "a chain is as strong as its weakest link" holds here, but the trouble is to find that weak link.

Let us consider, for example, a crankshaft. The majority of specifications state that the samples shall be taken, so many from one end of the shaft, so many from the other, at such-and-such a distance from the center. And what does this mean? If the metal were homogeneous and had been treated alike from end to end this would be satisfactory; but, unfortunately, in the manufacture of material of this kind the different parts are subject to different thermal and mechanical treatment, and even though we may refine by proper heat treatment, we cannot eliminate inherent imperfections.

Where, then, should the test be made? The most important point in any structure is the point at which the maximum stress will occur in practice, and manifestly it is at that point that the metallographic examination should be made. The metallic structure at that point may be entirely different from that at any other point, but a short examination will reveal this fact. It is rather amusing to note, for example in testing the shaft of a center-crank engine, that the test pieces taken as above will be at the point of minimum stress, while just outside the webs of the crank, where stress is greatest, no examination is made.

With castings, the proper location of the test specimen becomes more difficult, unless the material is to receive further treatment, and I believe that for small castings the most satisfactory method is actually to cast one or two extra pieces from the same melt for microscopic examination. In large castings, where possible, the coupon should be cast on the body of the

thicker parts, or if this is impossible, gated to the casting by wide gates. The part subject to greatest strain should be given the most attention.

The question might now be asked: What can the metallographer reveal that the chemist and mechanical engineer can not? Without going into detail we may say:

First, that his examination will check the chemical analysis and, in addition, will detect the condition in which the chemical elements exist; whether they are in the form required for the best results in service. He will also detect the presence of impurities of deleterious nature that the chemist will not find.

Secondly, he will detect unhomogeneous material.

Third, he will detect improper working, either too high or too low finishing or casting temperatures, or excessive cold work.

Fourth, he will in most cases detect whether or not the metal has been properly heat treated, and if not, suggest a treatment which will render the material satisfactory. In fact, if considering an average sample, he will predict with unbelievable accuracy the physical characteristics of the metal.

If he is examining a fracture, he may discover whether or not the failure was due to faulty material, faulty treatment, excessive stresses, shock or fatigue, and by sufficient investigation, like the pathologist, will finally indicate the exact point in the life history of the material where the trouble started.

The importance of this method is so great that the testing engineer must now take it up scientifically, or be relegated to the ranks of the "has beens."

Aside from the tests already mentioned, the suitability of a metal for special purposes is often determined microscopically or by special tests, as, for example, corrodibility in acids, in water, or gases; the electrical properties, as conductivity, resistance, permeability; the thermal qualities—specific heat, melting point, etc. For example, in type metal we need a fairly strong metal, of low melting point, which will expand on cooling; and the question is, what shall we select as an alloy, and, is the one we have selected the best for the

purpose? Again, in bearings we want a low-melting alloy, of low specific heat, which will expand on cooling, preferably; low coefficient of friction, and yet plastic. What metal shall we use to obtain the best results? Manifestly the engineer who draws up the specification should know, first of all, the characteristics of his material.

The logical method in design would be:

First, analyze the stresses carefully, both as to type and intensity.

Second, select the material to be used.

Third, draw up the specifications for the material, allowing the manufacturer as much latitude in methods of manufacture as is consistent with perfect safety.

Fourth, specify the physical tests, and be sure that the test specified brings out the exact quality of the material that you need to know to make the design rational throughout.

MACHINERY FOR CAPITAL VESSELS.

By Lieutenant-Commander H. C. Dinger, U. S. N., Member.

One of the naval lessons of the European war has been the need of adequate under-water protection to capital vessels. The need of such protection has not been fully appreciated in the past; but the deadly effect of the torpedo and mine, which has been indicated so often, has positively shown that under-water protection is one of the essential requirements in the design of capital vessels. In future designs this feature must be accorded full consideration.

Proper under-water protection and subdivision requires more compact machinery and a machinery layout that can be properly subdivided so that its parts may be placed in isolated compartments.

The requirements and the limitations of the machinery lay-out in respect to compactness and possibility of subdivision is one of the largest factors to consider in connection with securing proper under-water protection.

Aside from the space required, there is also the question to consider of the inter-connection necessary between the different machinery compartments. If the layout requires the presence of large and numerous steam and water pipes, ducts or other mechanical connections, the problem of a proper subdivision is much more difficult and complicated. It is desirable to have the machinery so that the different parts can be isolated in separate compartments or groups of compartments so that damage to one effects the rest as little as possible.

The essential matter in securing compactness of and lightness of the machinery layout is fuel economy. If one heat unit

can be made to do what previously required two, we will necessarily save weight and secure compactness. Hence, generally speaking, the most economical machinery will give us the lightest and most compact layout (fuel to be included). The demand for large cruising radius calls especially for fuel economy. Each heat unit and each unit of heating or of cooling surface must do the best possible.

An examination into the possibilities of a machinery layout for capital vessels that will take advantage of what is best and most promising in the new developments of engineering will disclose the possibilities of some interesting results and will indicate that marked improvement in economy and compactness in naval machinery plants is a practicable possibility.

THE CALL FOR SPEED.

The call for speed—speed greatly in excess of our wildest dreams of a few years ago—is becoming more and more urgent and insistent and this demand must be met by the engineering forces.

Accompanied with the demand for speed is the demand that it must be secured without sacrifice of the defensive and offensive qualities. This looks very much like a paradoxical demand, but it can be met to some extent by improved efficiency in engineering design and construction.

The requirements can be approximated by proper improvement in economy, by securing the best efficiency from every unit weight or unit surface that goes into the component parts of the machinery and by using those materials which have been found most suitable.

NEW POSSIBILITIES.

The most important of the late engineering possibilities that may be used to accomplish economy and compactness are the following: (1) The electric drive, (2) More economical turbines, (3) Mechanical reduction gears, (4) Superheat, (5)

Improved boiler baffling and water circulation, (6) Diesel oil engines, (7) Further development for conservation of heat lost by radiation and in the exhausts.

We may now inquire wherein the above matters will best serve us in securing better results than those to be secured in the designs now building.

(1) *The Electric Drive.*—The electric drive has come to stay, at least, for a while. Its full possibilities have not been realized in our present designs and considerable further improvement may be expected. The great advantage of the Electric Drive is its flexibility and the ability to isolate or segregate its various parts in separate compartments which require only a cable connection. It is in this particular that the electric drive enables us to give proper under-water protection. There is an absence of long shafts, very large and long steam or water pipes connecting several compartments and the operation of the plant is not adversely affected by completely isolating the various parts.

(2) *More economical turbines.*—Improvement in the performance of the Electric Drive rests mainly in providing more economical turbines and in using a higher degree of superheat than has been the practice.

The most economical turbo generator sets now known of are the Ljungstrom turbine sets. These were fully described in the August, 1916, number of the JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS by Lieutenant Carter, U. S. N.

These machines have now been operating with success for several years and are practically out of the experimental stage of development. Their use has been taken up by Beardmore, in England.

The Ljungstrom turbine with Electric Drive in large units will develop a S.H.P. at propeller on 8.5 pounds of water with steam at 600 degrees F., and possibly this result may even be improved upon.

The Ljungstrom turbine has therefore demonstrated a material gain in economy. It is claimed that other turbines can,

under special conditions, also approach this approved economy, but so far it has not been done. If it can be done by a turbine of less complicated construction than the Ljungstrom turbine it will be highly desirable to have such designs produced. Anyway the actual economy of the Ljungstrom system shows that material improvement is possible in steam economy for electric drive use.

(3) *Mechanical Reduction Gear.*—Turbine reduction gear has been generally successful. It will weigh slightly less, and in some cases give a better economy at full speed, than the present Electric-Drive installations.

It does not, however, have the flexibility nor does it present the facility for subdivision and isolation of parts that the Electric Drive possesses. It apparently finds its most suitable place on Destroyers and for plants of comparatively low powers where compactness is desired and where subdivision is not necessary.

(4) *Superheat.*—Superheat has been used in a more or less desultory fashion for naval work. Where it has been employed it has been successful. It has had to fight its way against prejudice, and in no naval or marine installation is the full practical gain due to its use fully realized. Superheat is especially desirable with oil-fuel boilers and in connection with high speed turbines such as are used in connection with geared and electric-drive installations.

In existing naval installations we have been at times working up to a temperature of 500 degrees F. A higher temperature can be successfully used. Temperature up to 600 degrees F. are successfully used in power plants, and there is no proper reason why the same should not be used on board ship. The use of superheat is preferable to going to excessive steam pressures. Pressures above 250 pounds per square inch are unnecessary. The limit of temperature permissible should be secured by superheat. High pressures add unnecessarily to weight of piping, drums, etc., and produce more difficulty in obtaining tight joints.

(5) *Improved Boiler Baffling and Circulation.*—Performances with oil-burning boilers give an efficiency of about 80 per cent. This performance can be improved by a more careful design of baffling and boiler clothing and a higher capacity per unit of heating surface can be secured.

The baffling should be arranged so as to secure the most efficient relation between the path of the gases and the circulation of the water, and also the area of the gas passes should be proportioned to the density. The velocity of the gases over the heating surface should be maintained as uniform as possible throughout the boiler. It is believed that some experimentation in the above field would develop boiler designs that could secure a boiler efficiency approaching 90 per cent. The principle of baffling arranged so as to maintain an even velocity of gases has been put to a practical test in a large power plant, and improvement both in economy and capacity have resulted.

An efficiency of only 80 per cent. should not be considered as the ultimate limit of boiler efficiency.

(6) *Diesel-Engine Generator.*—There has been special development in the practical working efficiency of Diesel engines, so that now Diesel engines which are thoroughly reliable are being built by numerous firms. Due to initial mistakes in design and construction and in the attempt to operate Diesel engines by inexperienced and untrained operators, difficulties were experienced which have created a strong and rather unreasonable prejudice against the Diesel engine for naval and marine work. There are limitations to the Diesel, and these are now fairly well known. For continuous operation in one direction, as for generator drive, the Diesel is entirely satisfactory within certain limiting cylinder diameters.

Up to certain limitations of cylinder size the Diesel is entirely suitable for direct-drive marine installations, and the very large number of actual successful installations bears ample testimony to this.

A good Diesel engine will secure twice as much power out of

a pound of fuel oil as will the most efficient steam plant. The Diesel will cost more, weigh somewhat more, but on the other hand, the operating force can be materially reduced.

(7) *Conservation of Waste Heat.*—This will consist in securing the greatest possible efficiency in condensers, feed heaters and in clothing and lagging material. There has been marked improvement in condensers and feed heaters, but the full use of the most efficient devices and systems is seldom used in actual plants. In the matter of exposure of heated surfaces and their lagging and clothing to prevent loss of heat by radiation there is great improvement possible.

The problem of clothing hot exposed surfaces has received little scientific thought and attention, and it is a field that can be improved.

The casings of oil-burning boilers must be very carefully constructed so as to avoid heat by radiation, and there is here a special field for the application of improved non-conducting clothing and lagging materials.

PROPOSED MACHINERY PLANT.

The following might be the basis for the machinery layout of a capital vessel.

Suppose the S.H.P. required is 35,000, we could have an electric drive installation as follows:

Two steam-driven generators of 10,000 kw. each;

Two Diesel-driven generators of 2,000 kw. each;

Four motors on four separate shafts.

Boilers.—Assuming that with improved boilers a boiler efficiency of 85 per cent. can be obtained and that the turbines with the electric drive are able to deliver a S.H.P. on 9 pounds of steam at 250 pounds pressure and at 550 degrees F. the steam can be supplied on about 24,000 square feet of heating surface, or about .8 square foot H.S. per S.H.P. This reduction of heating surface is to be secured by increased turbine economy and boiler efficiency and by increased average capacity of the boiler-heating surface.

This heating surface can be readily accommodated in four or six boilers, each of which can be placed in its own watertight compartments. About 15 per cent. of H.S. would be superheating surface.

Turbines.—The Ljungstrom turbine is at present the most economical turbine available for electric drive. There is no special reason to suppose that this turbine can not be successfully applied to units of 10,000 kw. capacity. No doubt other types of turbines can, by improvement, approach the economy that the Ljungstrom turbine has obtained, so that it is not unreasonable to expect that Turbo-electric installation of two 10,000 kw. generators can be secured that will give a S.H.P. on 9 pounds of steam at 250 pounds pressure and 550 degrees F. temperature. These steam-driven sets to work in conjunction with the Diesel-driven generators of 2,000 kw. each.

Diesel Generators.—The Diesel generators would be arranged to be operated when desired with the steam-driven sets at any power, including full power. The Diesel generators alone would be capable of driving the vessel up to a speed of 14 or 15 knots. The problem of paralleling the Diesel generators with the turbo-generators may present certain difficulties, but this problem may be solved.

The Diesel generators would produce a S.H.P. on about .43 pounds of oil, hence, at low speeds the fuel consumption would be less than half of what it would be with the most efficient steam-driven plant.

At the higher speed and at full power the Diesel units would improve the economy of the steam drive and at full speed would improve the economy about 8 per cent. So that the presence of the Diesel generators would increase the steaming radius at full power by 8 per cent.

The Diesel engines would also make possible other economies and present special advantages, viz:

(1) Diesel engine can supply hot feed water by using feed water for the jacket-cooling medium and by installation of feed heater in Diesel-engine exhaust.

(2) The vessel can be heated by utilization of Diesel-engine exhaust. These heating systems are successfully used on submarines and other Diesel-driven vessels.

(3) The air supply and compressors for the Diesel engine can be used for air salvaging purposes, thus making it unnecessary to duplicate the air-compressor plants now usually provided for such purposes.

(4) The regular ships' service dynamos can be reduced, as the cruising generators can supply desired electric current; if transformers are fitted, either D. C. or A. C. motors can be used for auxiliaries. Economical charging plants for charging batteries of submarines accompanying the fleet are provided.

The Diesel engine proposed for these generators is a very simple engine of the type that is very successfully used for power-house work. It would be free from many of the complications that have retarded the adoption of Diesel drive for vessels and is very easily operated and overhauled. Ample room for overhauling and a good lubrication system can be provided, so it will be free from most of the trouble that Diesel engines have experienced.

Diesel engines of the size and type required for this work have been built and are in successful operation and the details are fully developed, and the successful builders of Diesel engines are ready and prepared to supply Diesel-driven generators of the type required for use in connection with Electric-Drive apparatus.

ECONOMIC RESULTS OF PROPOSED LAYOUT.

At Full Speed.

Steam turbines, 9 pounds of water per S.H.P.

Boilers will give evaporation of 15 pounds of water per pound of oil, result equals .6 pound of oil per S.H.P.

Note.—This figure has been actually obtained in a small installation; in a much larger installation it can undoubtedly be considerably improved.

Diesel units, .43 pounds per S.H.P.

Combined results, .57 pounds per S.H.P.

The best promise of the present Electric-Drive installations is about .72 pound fuel per S.H.P.

There is therefore a possible improvement of 21 to 25 per cent. of fuel consumption at full power over our present steam-driven electric-drive installations.

At Cruising Speeds.

Diesel, .45 pound per S.H.P.

Electric, .9 pound per S.H.P. on the basis of present electric-drive machinery.

The cruising radius of the vessel can therefore be doubled by using the combination of steam turbo-generators and Diesel generators. There are at present no definite figures as to added expense, but this would be very slight, because the extra cost of the Diesel generators is more than made up by the reduction in cost of boiler-room equipment and possible reduction in ship's generating and air-compressing plant. The machinery weights for a 35,000 S.H.P. plant referred to will be about 2,200 tons. The estimated machinery weight for the *Tennessee* is 1,900 tons for 32,000 S.H.P.

The extra weight of the Diesel generators, 400 tons, is more than made up by the very material reduction in boiler weights and the saving in fuel required for full speed.

The weights of the combination plant would be made up approximately as follows:

Steam plant and auxiliaries 30,000 S.H.P., including motors for 35,000 S.H.P.	1,800 tons.
Diesel generators and accessories for 5,000 S.H.P.	400 tons.
Total	2,200 tons.

Should it be considered that paralleling Diesel generators and turbo-generators is not feasible, then another arrangement may be employed, giving better economy but requiring more weight for machinery.

Scheme B.

Total S.H.P., 35,000.

Two 4,500-S.H.P. Diesel generators.

Two 13,000-S.H.P. turbo-generators.

Four shafts.

Motors on wing shafts, 4,500 S.H.P.

Motors on center shafts, 13,000 S.H.P. each. This arrangement avoids any difficulty as to paralleling generators. Each generator can operate any one or more of the motors independently.

Results.

Full Power.

Steam, .6 pound per S.H.P.

Diesel, .43 pound per S.H.P.

Combined result equals .556 pound fuel oil per S.H.P.

Cruising Speeds.

Diesel, .45 pound per S.H.P.

Weights.

Steam plant and motors, 26,000 S.H.P. equal 1,700 tons.
Diesel plant and motors, 9,000 S.H.P., equal 800 tons.

2,500 tons.

The saving in fuel at full power is about 300 tons; hence, final weight is equal to weight of steam plant. Some additional space is required, but this is available on battleships where power is not in excess of 40,000 S.H.P.

Note.—The fuel consumption figures above stated are the best that the contemplated machinery should be expected to deliver, provided best use of devices to secure economy are made. It represents the possibility.

Details of Engines.

Two eight-cylinder double-acting four-cycle engines. Length of engine with generator, fifty feet; height, twenty-one feet; width of engine, nine feet three inches; revolutions, one hundred and seventy.

Such an installation can be accommodated in the spaces available for machinery with but slight alterations in the plans prepared for the steam turbo-driven electric drive.

There is still another possible arrangement that would utilize the economy of the Diesel engine.

Scheme C.

Total S.H.P., 35,000. Four shafts.

Two 13,000-S.H.P. turbo-generators with motors on inboard shafts.

Two 4,500-S.H.P. Diesel reversible engines directly connected to outboard shafts.

At low powers the inboard propellers would be permitted to revolve idly and the Diesel units would drive the vessel alone.

This arrangement would, perhaps, give a very good economy, but would not be as feasible as Scheme B for the following reasons:

The Diesel engines could not be used on anything but their own shafts.

The turbo-generators could not be used on inboard shafts.

Drag of idle propellers would be considerable.

The break-down of either Diesel unit would cut out one shaft.

The Diesel engine is not available as a charging plant or to supply electric power for any desired use.

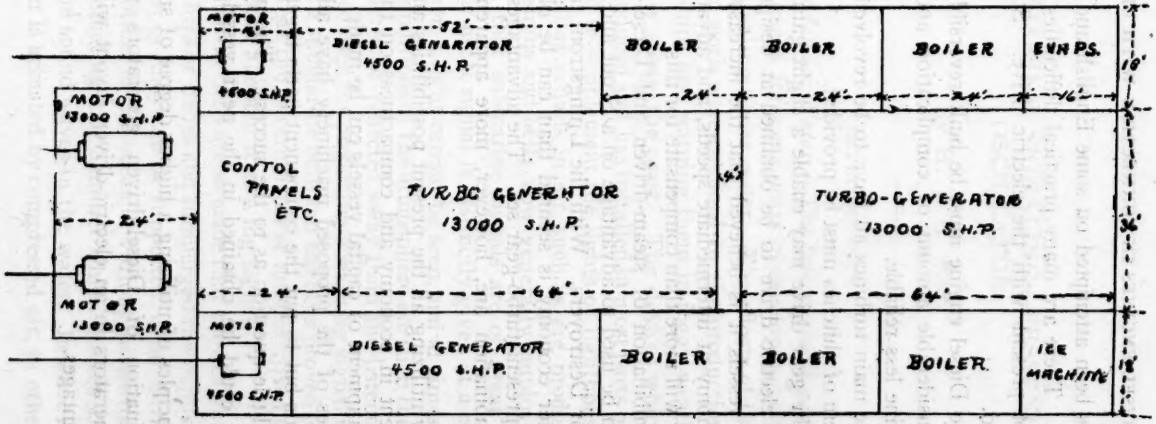
Scheme A is the most desirable, but the problem of regulation may not be readily met.

Scheme B presents no difficulties as to regulation and gives the best economy, but requires more weight and space.

Scheme B, considering all conditions and aspects of the problem, appears to be the most advantageous.

The over-all dimensions of compartments required by Scheme B are shown in sketch. This space is available in large battleship designs and is substantially the same as is required for the regular electric-drive installations that are being built.

Scale: 1 inch = 20 feet.



SCHEME B.—Proposed machinery compartments for battleship scheme B. Total S.H.P. 35,000. Two turbo-generators 13,000 S.H.P. each; two motors of corresponding power on center shafts. Two 4,500 S.H.P. Diesel generators; two motors of corresponding power on wing shafts.

USE OF DIESEL ENGINES IN CONNECTION WITH GEAR DRIVES.

This has been attempted on some English and German installations. There are many practical difficulties and complications not present with the electric drive. Some may be enumerated.

(1) The Diesel engine must be built reversible. This entails a considerable amount of complication and makes the Diesel engine less reliable.

(2) The main turbines all have to be revolved or an elaborate system of clutches must be provided.

While the gear drive may enable a slight gain in economy (over the electric drive to be obtained at designed power), for naval vessels it is believed that the increased flexibility, better economy at intermediate speeds, and advantages in arrangement will more than compensate for this.

The combination of steam-driven and Diesel-driven generators can be used to advantage on a Scout and may also be feasible for Destroyers. With the Ljungstrom turbine or its equal, better economy is secured than can be obtained with any other present turbo-gear set. The advantages of the proposed combination are, however, more apparent on larger vessels.

By fully utilizing all the present possibilities an exceptional improvement in economy and compactness on the latest machinery equipment of capital vessels can be made. The different features of the proposed machinery have all been tried, and it needs but to put the component parts together. There should be little question as to the success. The promised improvement could be obtained in the next vessels to be laid down.

The principles of utilizing a higher degree of superheat and the combination of Diesel-driven generators with steam-driven generators in an electric-driven layout will secure decided advantages.

SUBMARINE SIGNALING—FESSENDEN OSCILLATOR.

By H. J. H. Fay.

INTRODUCTION.

Signaling Under Water.—The Submarine Signal Company has been engaged for some years in developing and providing means of signaling under water. The efforts of the company in its earlier work were confined to the production of means of guiding ships and warning them in foggy weather of dangerous points along the coast. The devices which the company has produced and has installed in most parts of the world are designed to supplement the lights of lighthouses and gas buoys as well as sounds in air, such as are produced by the fog horns, sirens and bells now generally employed. It has been found that submarine signals are of great service in that by their use a signal is distinctly given at a considerable distance when a light cannot be seen, and at times and places where the conditions of the atmosphere are such that sounds in air are refracted or reflected away from the listener.

The device used for producing a warning signal under water is known as a submarine bell, and has been proved to be of such usefulness that the governments of most civilized countries of the world have adopted it as a fog signal.

The submarine bells are so made that they can be suspended from lightships or buoys and submerged the requisite distance under water, or they can be supported upon a tripod resting upon the bottom of the ocean. Various means have been devised for actuating the hammer which strikes the bell and produces the sound waves in the water. In some installations the hammer is actuated by compressed air, in others the blow

is struck by electrical means, and in others the motion of the waves is utilized to operate the striking mechanism.

The bell is the source of sound; that is to say, the source of the compressional waves which would produce sound in the ears of a listener if the ears could be submerged so as to hear the vibrations of the submarine bell. The use of human ears under water to detect these sound waves is clearly impracticable. Consequently, it became the problem of the Submarine Signal Company to provide—for each vessel using its signaling system—ears to receive the sound waves from the bell and to transform such sounds as the ship's ears received into sounds which would be audible to human ears.

This has been accomplished by mounting two tanks on the ship's plates, inside the vessel, one on each side, toward the bow. In each of these tanks is placed a microphone and the tanks are filled with water. Each microphone is so arranged that it can be connected by means of a switch with telephone receivers placed at the ears of the listener. When the waves from a submarine bell reach a ship equipped with ears of this kind, the sound waves agitate the sides of the ship and, through the water in the tanks, actuate the microphones, vary the electric current, and so produce a similar sound in the telephone receivers held at the ears of the observer. Clearly, the sound produced will be greatest when the telephones are connected with the microphone on the side of the ship nearest to the source of sound.

With an arrangement of this kind it is possible, regardless of fog or other weather conditions, not only to hear the warning signal given by the bell, but to determine the direction from which the sound comes. For if the bell tone is louder when the receivers are connected with the microphone in the port tank, the bell must be on the port side of the vessel; similarly, if a louder sound is heard when the receivers are connected with the microphone in the starboard tank, the bell must be on the starboard side. If, however, the sounds from the two microphones are equal, it is clear that the vessel is heading directly toward the bell.

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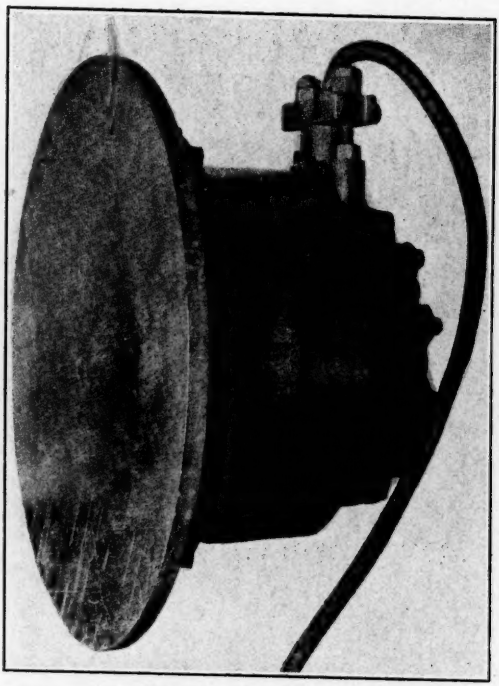
SUBMARINES AND ANOTHER SHIP MANEUVERING BY SUBMARINE SIGNALS.



FESSENDEN OSCILLATOR.

The submarine signaling system which has just been described affords ideal means of producing a fog signal. Not only is the sound produced sufficient to be heard at considerable distances, but the blows can be so timed that a code is given and thus indicate immediately to a listener the particular bell from which the signals originate.

The Submarine Signal Company has been endeavoring for the past few years to find means whereby the range of usefulness of submarine signaling can be extended, so that its apparatus can be used to produce not only fog or danger signals but such signals as will provide means of communication between the shore and a vessel, as well as between vessels either at anchor or moving rapidly through the water. With the signaling apparatus above described a vessel has means of hearing only. For communication the vessel must be provided—in addition to its receiving apparatus—with some means whereby quick and distinct signals can be produced and sent out from it through the water.

Bells such as have been described above have, in a number of instances in the past, been mounted upon submarine boats. It has been possible in this way for those on submarine boats to send signals to each other. Likewise signals could be sent and received between submarine boats thus equipped and mother ships having receiving apparatus and submarine bells suspended over their sides.

The bell method of communication did not prove entirely satisfactory in that the single bell stroke required a special code and lacked the speed which could be obtained with the dot-and-dash used in systems of telegraphy.

Other and more efficient means of communication had to be found and have been found in a new device known as the Fessenden oscillator.

DESCRIPTION OF OSCILLATOR.

The Fessenden oscillator surpasses the submarine bell both in power and in speed of operation. It can be readily mounted on shipboard, can produce signals equally effective when the

vessel is in motion or at rest, gives a signal of full power for any desired time (as contrasted with a bell, the vibrations of which lose their power rapidly), and can start or stop producing sound waves almost instantaneously. Added to the above advantages as a sound-producing device is the fact that it is a good sound-receiving mechanism, comparable in efficiency with the microphones now generally used in submarine signaling.

The practical advantages of a device of this kind need hardly be enumerated. By its use submarine boats can be maintained in constant communication with other vessels equipped with similar apparatus, regardless of whether the submarine boat is on the surface or submerged. By the use of the Fessenden oscillator the wireless apparatus on battleships is supplemented and means of communication provided in case the wireless masts of the ship are shot away. The Fessenden oscillator is likewise of value to merchant ships, as it provides means of communication in fogs, enables them to locate one another, and thereby reduces the dangers of collision. It is also of value in case of accident as a means of summoning aid and giving to rescuers the location of the ship.

Operation of Oscillator.—The purpose sought in designing the oscillator as a producer of sound was to provide means of pushing the water backward and forward with great rapidity. Speaking very generally, the larger the area of water pushed backward and forward the more powerful the sound; likewise, the greater the distance the water is pushed and pulled backward and forward the greater the sound.

In some respects the action of the oscillator is similar to that of the telephone receiver or the phonograph. In the telephone and phonograph a diaphragm is provided and so arranged that it is pulled backward and forward, thus compressing or rarefying the air in front of the diaphragm. In the oscillator there is a diaphragm which must be pulled backward and forward similarly, but as the diaphragm is in contact with water—an almost incompressible fluid—an enormously greater force must be used to produce its motion than is required in the telephone or phonograph, where only

air is to be compressed. Likewise, the diameter and thickness of the diaphragm of the oscillator are far greater than they are in the telephone.

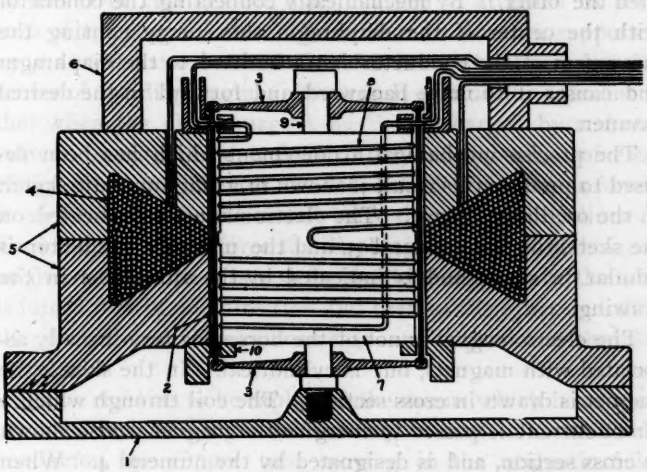


FIG. 1.

Fig. 1 is a diagrammatic sketch showing the working parts of the oscillator. In this sketch the diaphragm is marked 1. This diaphragm is about twenty-four inches in diameter and has a surface area more than two hundred times greater than that of the telephone diaphragm. The thickness of the diaphragm can be inferred from the sketch. It is apparent that great energy is required to bend this diaphragm in and out, especially when it is subjected to the pressure of the water in contact with its outer face. In what follows an attempt will be made to explain very briefly the means which have been devised for bending the diaphragm.

It is a well known fact that, if a current is passed through a wire and this wire is placed in a strong magnetic field, it is possible so as to form the wire and so to direct the current that the wire will move in the magnetic field. This is the general principle which has been utilized in the oscillator. A powerful electromagnet is provided and a powerful current is

passed through a conductor placed in the field of the magnet. The direction of the current is changed with great rapidity and the conductor is consequently forced first one way and then the other. By mechanically connecting the conductor with the center of the diaphragm the energy causing the movement of the conductor is transmitted to the diaphragm and causes it to move backward and forward in the desired manner.

The precise mechanical arrangement which has been devised to produce this result is shown in a diagrammatic sketch of the oscillator, Fig. 1. The electromagnet is indicated on the sketch by the numeral 5, and the movable conductor is tubular in shape and is indicated by the numeral 2 in the drawing.

The electromagnet is not of the horseshoe form usually associated with magnets, but is cylindrical. In the sketch the magnet is drawn in cross section. The coil through which a direct current is passed to energize the magnet is also shown in cross section, and is designated by the numeral 4. When direct current passes through the magnet coil 4, a north magnet pole is created around the inner surface of the magnet on one side of the magnet coil, and a south pole around the corresponding inner surface of the electromagnet on the other side of the magnet coil. The lines of force pass from one of these inner faces to the other. To increase the number of lines of force, or rather to reduce the reluctance of the magnetic circuit, an iron core, marked 7, is placed within the magnet so that the lines of force pass from one pole through a very short air gap to this core, which is cylindrical in shape, through it, and back to the other pole piece through a similar short air gap. This peculiarly shaped magnet with its core is one essential feature of the oscillator.

The other essential feature of the oscillator is the conductor which moves in the magnetic field existing in the air in the narrow gap between the pole pieces and the core. This conductor is the copper tube 2, which touches neither the pole pieces nor the core, but is capable of movement in the direc-

tion of its axis backward and forward in the magnetic field. This conductor—this copper tube—is so held that it can have motion only in the direction of its axis. This is accomplished by means of two flexible discs, marked 3, attached to both ends of the copper tube. These discs are in turn supported upon a steel rod 9, one end of which is rigidly attached to the center of the diaphragm. The result of this arrangement is that when the electromagnet has been charged by a direct current passing through field coil 4, and a current of electricity is passed through the copper tube, the tube is forced to move in one direction and thus, through the medium of discs and the rod, to force the diaphragm in one direction. If an alternating current is passed around the copper tube, the tube is forced first in one direction and then in the other and thus the desired to-and-fro movements are imparted to the diaphragm.

There remains to be explained the means which have been provided for obtaining an alternating current in the movable conductor, the copper tube moving in the magnetic field. Upon the core 7, are wound many turns of wire 8. When an alternating current is passed through these turns of wire, currents of electricity are created by induction in the copper tube. The currents of electricity thus induced in the tube are of great strength, owing to the fact that the copper tube is of low resistance and is in close proximity to the wires carrying the inducing current. It is to be noted that the lines of force are in one direction when passing from one pole to the core, and in the other direction when passing from the core to the other pole of the magnet. As a consequence of this, the direction of the windings on the core must be reversed so that all of the current induced in the copper tube may be effective in forcing the tube in one direction or the other. If the windings upon the core were not reversed, as shown in Fig. 1, the forces exerted upon the tube would neutralize each other, the tube would not move, and the oscillator would not operate.

Oscillator Sending.—It will be apparent from the above that in order to create compressional waves under water the

field magnets must be energized by a direct current passing through the magnet coils, and this direct current must be passing continuously through this coil so long as the oscillator is in use.

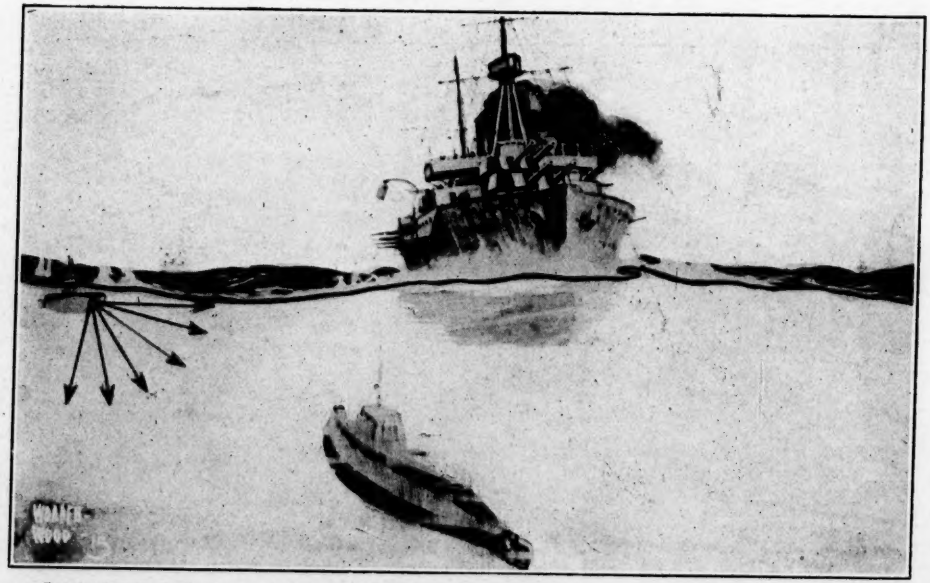
The current actually used to energize the field magnets has a strength of seven and one-half ampères.

The diaphragm is set in motion by passing an alternating current through the windings 8 on the core of the oscillator. This current induces currents in the copper tube and the motion of the tube is imparted to the diaphragm by means of the discs and rods. The alternating current usually employed has a strength of eleven ampères and is obtained from a source of one hundred and eighty volts.

The compressional waves thus produced in the water by the motion of the diaphragm are virtually sound waves and they spread out from the oscillator to very considerable distances. If a receiving device, like the microphone above described or the oscillator itself when arranged for receiving, is placed in the water so that these waves strike it, these water waves can be again transformed into electrical waves which can be made audible by means of a telephone receiver.

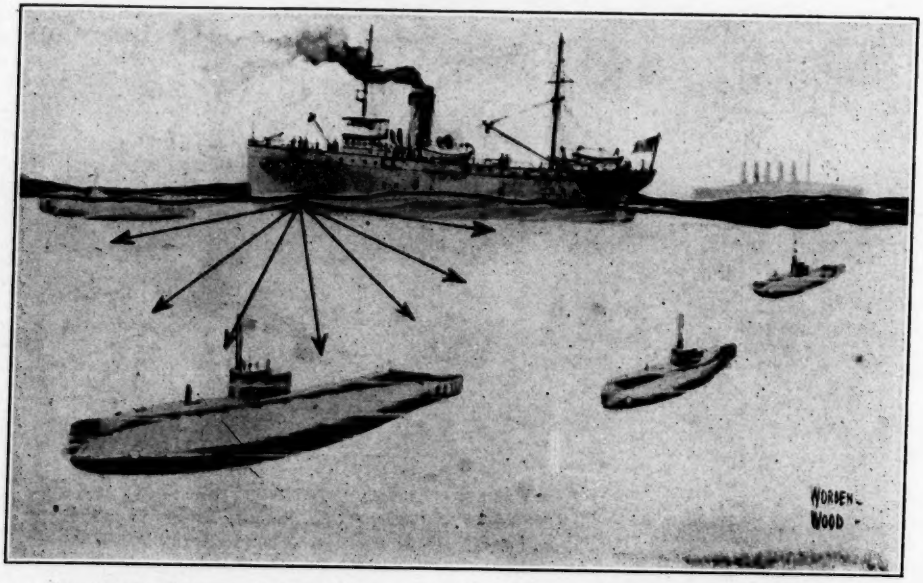
Oscillator Receiving.—The oscillator can be arranged for receiving by removing the source of alternating current and substituting one or more telephone receivers. The operation of the oscillator as a receiver is then simply a reversal of its operation as a sender. The compressional waves in water force the diaphragm of the oscillator in and out; this moves the copper tube backward and forward across the lines of magnetic force; these movements of the tube induce currents in the windings upon the core, and the currents thus generated actuate the telephone receiver; in this way sound waves in water are transformed into audible sound waves in air.

One of the principal difficulties in receiving submarine signals is that the receiving devices may be actuated by water waves other than those produced by the signal-sending mechanisms. Thus the splashing of the surface water waves upon the sides of a vessel will produce motion within the sub-



SUBMARINE ASTERN OF RUNNING BATTLESHIP GIVING RANGE TO WAITING SUBMERGED SUBMARINE IN FOREGROUND.

AMERICAN SOCIETY NAVAL ENGINEERS



SUBMARINE TENDER COMMUNICATING WITH SUBMERGED MANEUVERING SUBMARINES.

marine receiving apparatus and thus create noises foreign to those which it is wished to hear. Likewise the jarring produced by the motion of machinery may give rise to disturbing sounds. These foreign noises may be so loud in some cases as to obliterate the submarine signals.

The oscillator is less sensitive to these foreign noises than other submarine receiving devices; yet, on the other hand, it reproduces the submarine signals with great clearness and with constant uniformity.

Character of Sound.—The waves produced by the oscillator follow each other with the frequency of the alternating current used to actuate it. The frequency with which the oscillator has been designed to operate is 540 cycles per second; that is to say, the diaphragm is forced in and out 540 times in a single second. All parts of the oscillator mechanism have been given such dimensions as will be effective in producing water waves of this periodicity.

Range.—The art of submarine signaling is as yet too new to make it possible to give accurate figures as to the distances to which the signals from the oscillator are audible. The greatest recorded distance to date is about thirty miles and the average distance somewhere between five and ten miles. Nothing is as yet known definitely of the causes of the variations in the distances to which submarine signals can be transmitted. These variations have been attributed to differences in the ocean's bottom, to the position of the oscillator, and to other conditions not yet fully understood.

Signaling.—The oscillator as above described will produce a series of waves of constant frequency or pitch so long as the alternating current is passed through the coil wound on the oscillator core. By applying the alternating current to the oscillator for different lengths of time it is possible to send series of waves of greater or less duration so that such series will correspond to the dots and dashes used in communicating by the Morse code.

The speed of signaling with the oscillator depends upon the skill of the operator, and is practically the same as that ob-

tained in wireless telegraphy. It is not at all unusual in submarine telegraphy to send messages at the rate of twenty words a minute.

A few of the mechanical features of the oscillator may be briefly described.

Dimensions of Oscillator.—Diameter of oscillator diaphragm, about 30 inches; its overall depth about 16 inches; its weight about 1,200 pounds.

LOCATION OF OSCILLATOR APPARATUS.

Location of Oscillator on Shipboard.—The position of oscillators on shipboard is similar to that of tanks containing the microphones described in the introduction. Two oscillators are used in each vessel, one being placed on each side not far from the bow. This arrangement permits signals to be sent from either side of the vessel, as may be desired, and likewise makes it possible to find the direction of the source of an incoming signal in precisely the same manner that it is possible to determine, by the use of tank microphones, the direction from which the signals of a submarine bell are heard.

Location of Oscillator on Submarine Boats.—It has been found that the best position for oscillators used on submarine boats is, as a rule, in the forward ballast tanks. The oscillators should be sufficiently deep in the boat to enable signals to be sent and received when the boat is on the surface. Also, the oscillators should not be placed so far forward in the bow that they will operate upon the broken water produced by the motion of the boat through the water. The most powerful signals can be produced when the diaphragm of the oscillator is covered by as dense water as possible.

Location of Oscillators on Ships.—The best position for oscillators on ships varies with the conditions found to exist upon the particular ship which it is desired to equip. It has been found well to have the oscillators about 15 feet below the water line if possible, but not so near the keel that the compressional wave can travel from one side to the other,

thereby interfering with direction finding. It is also necessary to have the oscillators near enough to the bow to face slightly forward.

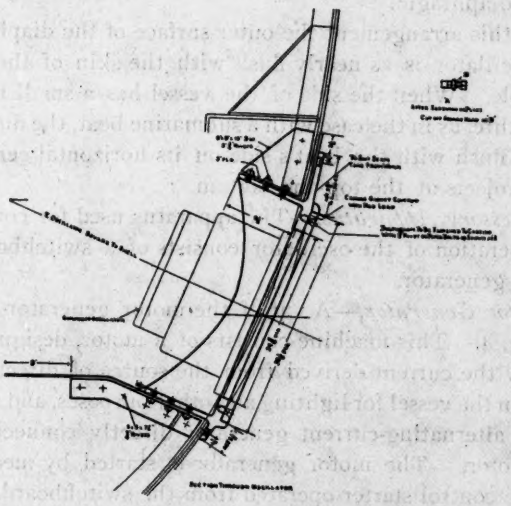


FIGURE 2

Method of Mounting Oscillators.—The method today employed in mounting an oscillator is to substitute the face of the diaphragm of the oscillator for the skin of the ship. The mechanical means of accomplishing this are indicated in the detail drawing shown as Fig. 2. The method of mounting will be briefly described.

A cast-steel supporting ring is required, this ring being of the size and shape shown in Fig. 2. This ring is mounted in a hole cut in the side of the ship and securely riveted to the plates of the ship. As will be noted by reference to Fig. 2, all sharp corners or edges of this ring which project into the water are rounded so that the water will be disturbed as little as possible in the vicinity of the oscillator.

The oscillator is inserted into this ring so that the face of the diaphragm is flush with the outer edge of the ring. The

position of the oscillator is clearly shown by the light lines in Fig. 2. The oscillator is attached to the ring by means of rivets or flush countersunk tap bolts passing into the flange of its diaphragm.

By this arrangement the outer surface of the diaphragm of the oscillator is as nearly flush with the skin of the ship as possible. When the side of the vessel has a small radius of curvature, as in the case with a submarine boat, the diaphragm is put flush with the boat's side on its horizontal center line and projects at the top and bottom.

Accessory Apparatus.—The apparatus used for controlling the operation of the oscillator consists of a switchboard and motor generator.

Motor Generator.—A cut of the motor generator is given in Fig. 4. This machine consists of a motor, designed to be run by the current derived from the source of direct current used on the vessel for lighting and other purposes, and a single-phase alternating-current generator directly connected with the motor. The motor generator is started by means of a remote control starter operated from the switchboard.

Switchboard.—A cut of the switchboard is shown in Fig. 3. On the switchboard are instrumeuts, switches, rheostats, sending key and telephone receivers. The instruments consist of a voltmeter, ammeter and frequency meter by means of which the operator can ascertain when the required amount of power at the correct frequency is obtained. The best point at which to operate the oscillator has been found at a voltage of 180 volts and 11 ampères at its own frequency of 540 cycles. The switches consist of a send and receiver switch, an off and on switch for oscillator, oscillator field switches and motor generator starting switches. By means of the send and receive switch and the off and on switches the operator can quickly shift from sending to receiving and from one oscillator to the other, thereby insuring speed of communication. The rheostats are for use in setting the motor generator to give the required voltage and frequency.

The Fessenden oscillator, with its associated apparatus,

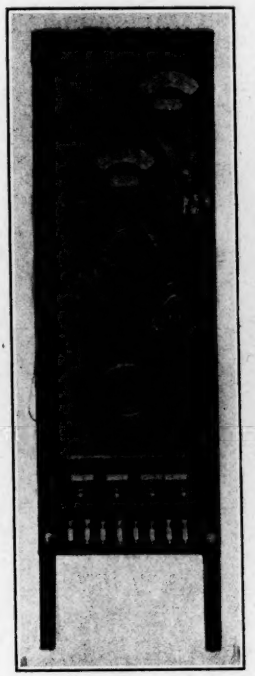


FIG. 3.

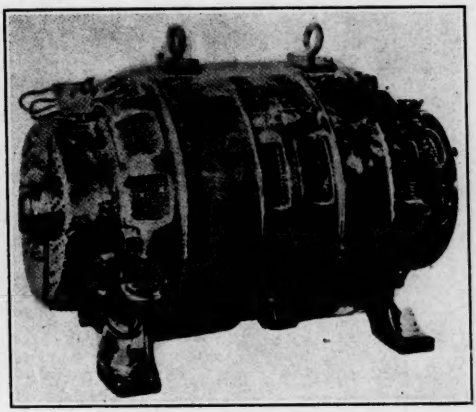


FIG. 4.

thus provides simple and efficient means of rapid communication. It is likewise a sound-receiving mechanism capable of use in finding the direction from which the waves produced by a bell or another oscillator are coming. The Fessenden apparatus can be transformed, by the use of a simple switchboard, from an extremely sensitive sound-receiving device into a producer of compressional water or sound waves of far greater amplitude or strength than have been heretofore obtained.

When the great energy required to create such submarine waves is considered, the speed and ease with which they can be produced and stopped with the Fessenden apparatus is very remarkable.

DISCUSSION OF PAPER ON STRENGTH OF BOILER FURNACES BY JOHN AIREY.

(Published in August, 1916, JOURNAL.)

By H. J. VANDEREB, HARTFORD, CONN.

In the August number of the JOURNAL I have read with considerable interest an article by Professor John Airey, the purpose of which apparently was to expose the inadequacy of the flue formulae, as regards safety, of the A. S. M. E. Boiler Code by comparing the results of these formulae with some other flue formulae promulgated by various authorities.

As the writer of this letter has had the advantage of figuring allowable pressures on a very large number of flues covering practically all varieties of sizes actually in use, and also knows from close personal touch with accident statistics in this country that flue failures due to structural weakness as known by theoretical determination contribute a microscopically small proportion of the total number of flues collapsed, he is quite certain that Professor Airey's criticism of the A. S. M. E. Code formulae is not warranted.

In fact it seems entirely misplaced, since, judging from what still is actual practice in this country as regards allowing pressures on flues, the A. S. M. E. Code formulae appear ultra conservative, and their prime virtue is that they are consistent with the most up-to-date and thorough research on this phase of mechanics. Professor Airey's reference to "the danger" of the A. S. M. E. Code formulae would therefore seem almost humorous, especially when it is considered that flue failures, aside from those cases of flues that come down due to overheating in connection with excessive amounts of oil entering boilers, are very rare.

It would further appear that Professor Airey's observations

of the pressures allowed by other authorities are not correct in several instances.

Taking up Professor Airey's comment on some of the various rules quoted, I would call attention to the fact that Fairbairn's rule is based on experiments with wrought-iron tubes that were lap riveted and brazed, so that further allowance for the weakening effect due to a riveted joint is obviously not necessary, especially when making comparisons.

This rule is, however, inconsistent in that it permits of pressure determination on flues of any length and produces pressures for long flues that are very low and for the shorter flues excessively high.

The fact has been quite definitely established that for lengths of flues over about six times the diameter, any increase in length for a given diameter does not materially decrease the collapsing pressure.

By a series of tests, conducted on a most exhaustive basis, Professor R. T. Stewart found this formula for some conditions to err as much as 400 per cent. (Transactions A. S. M. E., 1906, Vol. 27.)

In the book written by Hutton, no mention is made of the fact whether the flues tested, on which he based his formula, were riveted or welded, but, judging from their size and the time this took place, it is fairly safe to assume that they were riveted flues, so that the weakening effect of a joint does not have to be separately taken into account, as it certainly never is in practice.

Hutton's formula has the same faulty characteristics as that of Fairbairn, which has led to the use of the length of riveted sections for "L" in the formula instead of the total length of flues that are made up in sections by means of lap-riveted girth seams.

By doing this, an allowable working pressure that seems more reasonable, and which sometimes may still be considered safe, is obtained on long flues.

Plain furnace flues of the length given in the concrete case submitted by Professor Airey, namely, 94 inches, were rarely

made in one length, so that a large majority of the flues at present in use have pressures allowed on them as based on this practice of taking the length of sections for "L." In some cases the pressure resulting appears unduly high.

This practice of using the length of sections for "L" is still followed more or less to this day in the manufacture of inferior grades of boilers and other pressure vessels.

Needless to say, the code formula gives pressure results considerably lower than the values obtained in the manner just mentioned with Hutton's rule.

The German Government rule is not quoted by Professor Airey in its entirety. This rule reads as follows:

$$t = K \times \sqrt{P \times L \times D} + 0.1\,c.M.$$

The constant 0.1 is for boilers in river steamers. For boilers in sea vessels this constant is 0.2. The extra thickness resulting from this augmented constant is required to offset the stronger corrosive action in boilers on board sea vessels due to the possibility of salt water entering them. Quite obviously, for the purpose of making a comparison with the requirements of the A. S. M. E. Code, which is intended for stationary practice, it would appear that 0.1 is the proper constant to use instead of 0.2, as was done by Professor Airey. On the flue mentioned in the concrete example (29 inches diameter, 94 inches long, $\frac{1}{2}$ inch thick) by using the constant 0.1 the pressure becomes 90 pounds against 75 pounds as figured by Professor Airey.

Lloyd's rule, $P = \frac{89{,}600 \times t^2}{L \times D}$, used to be quite generally used in the United States for riveted flues, but is now more or less in the discard. Wherever it is still prescribed, as, for instance, in the boiler rules of the City of Philadelphia and other rules that were designed to cover the early practice of boiler construction, the length of the lap-riveted sections may be taken for "L," which in some cases has led to absurdly high-pressure allowances.

In the practice of applying Lloyd's rule it was never cus-

tomary to make a special allowance for the character of the longitudinal seam, speaking with regard to American practice at least, and this certainly is not done now in localities where this rule is still used.

The U. S. Government rule (Steamboat Inspection Service) is not identical with Lloyd's rule, as stated by Professor Airey, the latter having been abandoned about 10 years ago. The present U. S. Government rule is

$$P = \frac{51.5}{D}\left((18.75 \times T) - (1.03 \times L)\right).$$

This is the same formula as given in the A. S. M. E. Code for flues of which the length is 120 times the thickness of the place or less.

According to this formula the pressure on the flue of the concrete example would figure 94 pounds safe working pressure. There is no stipulation in connection with this formula regarding the weakening effect of longitudinal seams, and consequently the pressures on all boilers in steam vessels where this rule happens to be the deciding factor are never determined making such allowance, and it may be said in this connection that to (quote the Government Rules) such "plain circular riveted flues, furnaces and cone tops" are quite generally lap riveted.

The fact that the U. S. Government rule allows for "L" in the formula to be taken the length of lap-riveted sections causes pressures to be used that appear entirely out of proportion in some cases of very long, thin flues.

The British Board of Trade rule is not correctly quoted by Professor Airey. He quotes it as:

$$P = \frac{75{,}000\ t^2}{(L + 1)\ D}.$$

The constant in this formula may be taken at 77,000 for steel flues with single-riveted lap seams. This brings the pressure on the flue in the concrete example up to 75 pounds.

There is no provision that the length of riveted sections may be taken for "L" in this formula, but there is a stipulation limiting the pressure on plain flues to $\frac{9{,}900 \times t}{D}$ pounds.

Since the British Board of Trade allow various constants in the flue formula relative to the type of joint it is, of course, not necessary for Professor Airey to make further allowances for the weakening effect of a single-riveted lap seam, as he did, as this is taken care of by the constant.

Summarizing the results of the various rules taken up here in the light of their practical application as stated, we get:

SUMMARY.

	Pounds, W.P.
Fairbairn rule,	155
Modified Fairbairn rule,	125
Hutton's rule,	120
German Government rule,	90
Lloyd's rule,	99
U. S. Government rule,	94
Longridge rule,	157
British Board of Trade,	75
Average,	114
A. S. M. E. Code,	99.7

Of course, as pointed out in the foregoing, the fact that some of the above mentioned rules allow the length of lap-riveted sections to be used for "L," in the case of flues so constructed, changes the values of pressure as given considerably, if, for instance, the flue of the concrete case under discussion were made in three sections that were joined by single-riveted lap-girth seams, we would get allowable working pressures as follows:

	Pounds, W.P.
Hutton's rule,	208
Lloyd's rule,	296
U. S. Government rule,	209
Average,	238
A. S. M. E. Code,	99

It is particularly in this feature of taking for "L" in the various formulae the length of riveted sections that the Code rules are a marked improvement, as this is specifically prohibited for the Code formula. (See interpretation by Code Committee in Case No. 22.)

It is extremely doubtful whether in the majority of cases one is justified to consider the reinforcing effect of a single-riveted lap seam connecting the sections of a riveted flue, sufficient to disregard the total length of the flue and simply consider the greatest length as that of one section, the same as is done when the sections of a plain furnace flue are joined by the Adamson ring construction or similar reinforcement, especially so when such flues built in lap-riveted sections are quite long, say 5 or 6 times the diameter.

A few examples of actual flues may serve to further show the conservative amount of pressure allowed by the A. S. M. E. Code formula and what it has heretofore been quite generally the practice to allow, by taking for "L" the length of sections, according to Hutton's, Lloyd's and the U. S. Government rules.

58½-in. length, 2 sections, 26-in. diam., ⅜-in. plate:

	Pounds, W.P.
Hutton's rule,	135
Lloyd's rule,	198
U. S. Government rule,	163
Average,	165
A. S. M. E. Code rule,	103

88½-in. length, 3 sections, 24-in. diam., ⅜-in. plate:

	Pounds, W.P.
Hutton's rule,	146
Lloyd's rule,	213
U. S. Government rule,	176
Average,	178
A. S. M. E. Code,	72

100½-in. length, 3 sections, 24-in. diam., ⅝-in. plate:

	Pounds, W.P.
Hutton's rule,	137
Lloyd's rule,	188
U. S. Government rule,	167
Average,	164
A. S. M. E. Code,	63

NOTES.

HIGH-PRESSURE STEAM STOP VALVES.

THE DESIGN AND CONSTRUCTION OF HIGH-PRESSURE STEAM STOP VALVES.*

By D. MacNicoll (Member).

In designing high-pressure steam stop valves the essentials to be observed are: Tightness, reliability, simplicity, and safety. In small sizes not exceeding 4 inches or thereabouts and of single-beat type these essentials are obtained without much call for special consideration, but as the sizes increase the design demands careful attention. In the succeeding columns an investigation of the various designs is contributed, and prominence given to details which affect what might be termed the general efficiency and safety of the fitting.

Fig. 1 shows a type of single-beat unbalanced stop valve frequently used as a boiler stop. The same design in more or less modified form is used throughout most main steam-pipe systems, being met with as an intermediate shut-off or bulkhead stop valve. The valve shown is of 8 inches diameter, the material of chest being cast-iron, internal parts gunmetal, and the working pressure 200 pounds per square inch. In many instances chests of this size and for this pressure are made in cast steel, but provided the design is carefully observed the choice of material is optional. For larger sizes or higher pressures cast steel is desirable. Independent of pressure, wherever superheat is present, cast steel is essential, and the internal fittings in this instance should be of a high-tension bronze, preferably a nickel alloy.

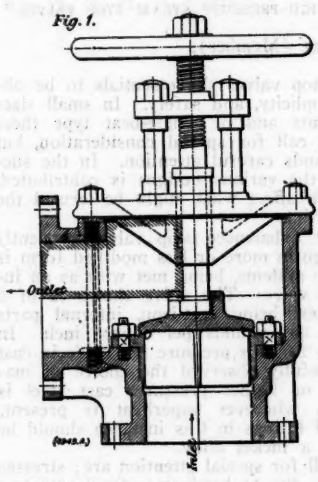
The features in the design which call for special attention are; stresses in the chest due to pressure; stresses due to hardening down; stresses due to expansion of spindle with valve shut; tendency of valve to rotate when steam is flowing through; liability to fracture of seat studs. In settling the thickness of body metal the general method is to insure that the aggregate section is such as to allow of a safe stress per square inch of section, but on investigation it is readily admitted that all parts are not equally stressed.

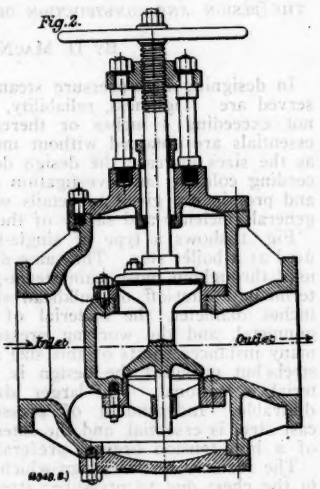
In Fig. 1, assuming that the pressure is present throughout the chest and acting at right angles to the plane of section shown, the stress on the part lettered A is much higher than on the other portions. By way of illustration:—The pressure on the upper half of the outlet and half the cover is approximately 8,950 pounds, which stresses the section on the top left-hand side to 1,061 pounds per square inch, whereas the average stress for the aggregate section is only 891 pounds. The cover, of course, takes a portion of the stress due to frictional resistance, and the tighter the cover nuts are screwed up the greater the assistance given to the weaker section. In a sense this is "robbing Peter to pay Paul," as it simply puts up the stress on the studs.

The stresses resulting from excessive hardening down are in many instances exceptionally high, as is often proved by the burst condition of valve lids. This hardening down is due to an attempt to overcome

*Paper read before the Institute of Marine Engineers, December 12, 1916.

leakage. It appears providential that the valve lid gives way in nearly every case, or otherwise there might be far more fatal accidents in this connection. Steel crossheads and turned pillars should always be fitted in preference to cast bridges, the latter being anything but safe, particularly in the larger sizes. There is no doubt that in valves over 4 inches diameter a great benefit accrues from the adoption of a flexible seating in conjunction with the solid face. This was introduced by Messrs. R. R. Bevis, J. H. Gibson and Cockburns, Limited. With this arrangement leakage from distortion is obviated, and in addition there are two faces, so that if one is leaking from any cause outside of distortion there is always the other one to fall back upon.





Another source of high stress is when the valve has been shut down hard when cold and afterwards steam from other boilers admitted to the top of the body. Even with steam in the inlet side only a fair degree of heat is transmitted to the spindle. In each case the subsequent tendency to expansion puts a stress on all the parts. An actual experience of the authors is illustrated in Fig. 2. This was a double-beat type of valve and the bottom seat was attached as shown. The valve was hardened down for a steam test, with the result that after the test a number of the manganese bronze studs from the bottom seat were found lying broken in the bottom of the chest, and all showed signs of fracture, the expansion of the spindle having caused this. Apart from this, the design of seat is not to be recommended even if the studs do not give way, as the tendency thereby is to cause a leaky joint between chest and seat. Seats should be arranged where valves have to be hardened down so that this hardening tends to make the joint tighter, that is to say the gunmetal seat should be upon the top of the steel seat, and not below it as shown. One method of counteracting the damage from expansion is by fitting a spring crosshead. This is shown in Fig. 3, and is loaded generally from 1¼ to 1½ times the boiler pressure per square inch of valve area, and a higher load than this cannot be put upon the valve spindle, no matter what purchase is used. In Fig. 1 stays are sometimes

fitted as shown dotted. These only serve to stiffen the cover and, as will be noticed, still further reduce the section of metal at an already weak section.

The rotation of winged valves when steam is flowing is due either to a slight angle on the wings or to eddies in the steam. The valve can be prevented from rotating, but to do this a pin must be fitted or other means adopted. The damage done to valves from this rotation is hardly credible, and calls for the adoption generally of pintle or centre-guided valves.

Fig. 3.

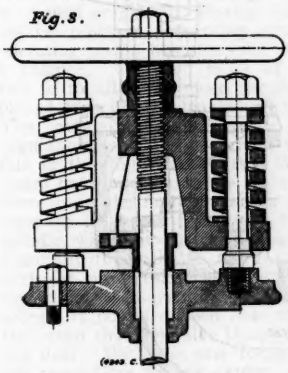
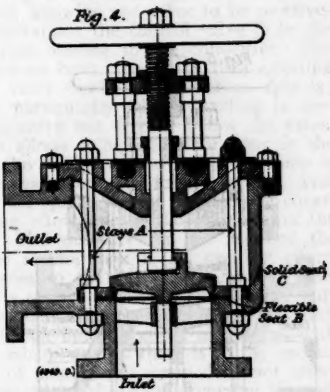


Fig. 4.



The source of much trouble is the fracture of seat-securing pins or studs. In the large sizes it is quite inadmissible to drive the seats in and secure them by means of a pin or pins through the side of the chest and it is absolutely prohibited in all sizes when superheated steam is present, experience showing in this case that seats driven in and secured by side pins become quite slack and allow leakage to occur. The seats in larger sizes are therefore secured in the manner shown in Fig. 1. The only way to avoid serious trouble with this design is to run a wire through the heads of all pins or through split pin holes, if studs are fitted. This prevents the fractured portions being carried to some part where serious damage may be caused.

The question of draining generally cannot receive too much attention, and means should always be provided for efficiently getting rid of all water in both inlet and outlet before a valve is opened.

In Fig. 4 a type of valve is shown which it is suggested is an improvement on that shown in Fig. 1. The first feature that attention is desired to be drawn to is the method employed so that the cover may take up a share of the bursting pressure which acts at right angles to the axis of the spindle. This consists of a tapered spigot on the chest with a corresponding tapered recess in the cover, making a metal to metal joint. The stresses from hardening down and expansion are mainly taken up by the four stays A, which also serve to secure the seatings. With this arrangement no loose parts are situated inside the chest, with the consequent prevention of damage from this source. The valve is centrally guided, thus preventing rotation. A flexible seating is shown at B and when this and the corresponding valve face engage, the solid seat C and its corresponding valve face are 1/100-inch open, so that when this latter seat and face engage the flexible seating is deflected 1/100-inch. Fig. 5 shows the same design of valve as in Fig. 4, but of straight

through or globe type; the design calls for the same attention as in the previous case.

BALANCED VALVES.

In single-beat valves of sizes over 4-inch and pressures in the region of 180 pounds and above, the valves are difficult to operate unless large diameter handwheels are fitted or handwheel spanners used to give the

Fig. 5.

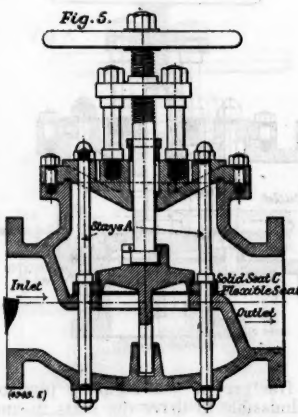
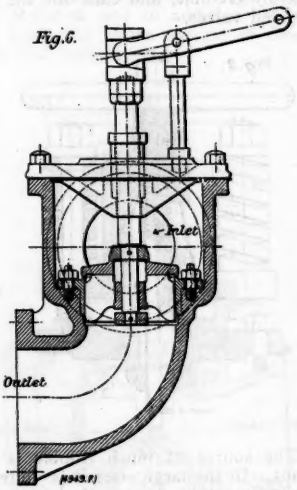


Fig. 6.



same effect or gearing introduced. This difficulty is due to frictional resistance in the screw and not to the direct load which, but for that resistance, would allow for quite a reasonably sized handwheel and comparatively quick-pitched thread in valves up to about 12-inch diameter. For regulating purposes large diameter handwheels or gearing are objectionable, not enabling the valve to be handled as expeditiously as is desirable. A compromise for this type of valve is effected and is shown in Fig. 6. A pilot valve is introduced and lifts first, balancing to a certain point the pressure in inlet and outlet branches. The main valve is then lifted against much less load. This fitting is extensively adopted, but occasionally trouble is experienced from the main valve pulsating or clattering, in reciprocating machinery, due to the fluctuating flow of steam—which causes damage to the internal parts. Solid double-beat valves got over this trouble in the majority of cases and were perfectly balanced to all practical purposes, but they had one outstanding defect—they were rarely, if ever, steamtight, which was very unsatisfactory. For reciprocating work this did not affect the steam consumption to any extent, as it was only when the machinery was at a standstill that leakage was occurring, and if a stoppage was of any length of time the boiler stop valves could be shut down. For turbine machinery, however, leaky astern valves can very seriously affect the economy. The leakage in the solid double-beat stop valve was due to differences of expansion between the double-beat valve and the seatings in the chest and could not be got over. A solution of the difficulty was ultimately found in introducing a flexible disc to one of the beats, and the names associated with this introduction were also Messrs. Bevis, Gibson, and Cockburns,

Limited. This valve has been very extensively adopted in varying designs and is exclusively used for all regulating valves in His Majesty's Navy.

Fig. 7 shows a double-beat type of valve used for reciprocating work for regulating purposes and also as a throttle valve, doing away with the butterfly type. In this latter capacity a very light duty is thrown on any form of governing gear, while in larger sizes it obviates the necessity for employing a separate throttle engine, say of the Brown type. For ordinary opening and closing as a regulating valve the pin B is locked by the detents as shown, thus allowing the valve to be positively opened or closed. During these operations the control valve is in the neutral position, preventing steam from passing to the condenser. The operating piston has also inlet pressure on both sides of it, thus effecting a balance. The two beats of the valve have saw-formation spigots; this is to allow of close regulation, particularly where shafting is employed to bring the handwheel to a position not directly below the valve. The back-lash in this instance often allows a big jump to occur in the regulation, which is prevented with the design as shown. This jump is due to the following cause: With a valve of this design, that is, with steam entering outside the beats and the spindle going through the cover, there is in the first instance a closing effort due to the pressure on the increase in area of the top beat. The pressure on the area of the spindle where it passes through the stuffing box is an opening effort. Generally the ratio of the former area to the latter is as two to one, with the result that when the valve is opened to an extent allowing half the initial pressure to be present in the outlet the efforts balance each other. A pound or two over this and an opening effort is present, with the result that the valve jumps open whatever play there is in the operating gear. With the saw formation of spigots this jump does not affect the revolutions to any extent. If instead of the spindle coming through the top of the cover it had come through the bottom of the chest the effort would have been a constant closing one, and there would be no necessity for the provision of the saw formation of spigot. Further on, valves with their spindles coming through the bottom of the chest are shown in Figs 7*d*, 8, 9 and 10. The saw formation of spigot is a refinement to give close regulation from dead slow to full speed, as many double-beat valves are made without this formation. When the valve is opened to any degree of regulation it can be shut and opened again by the throttle gear in the following manner: In Fig. 7*a*, *b*, *c* the throttle lever B is moved from position 7*a* to 7*c*. The control valve during this operation moves through position 7*b*, holding the main valve open till the detents have cleared themselves from the pin B, then with the throttle lever in position 7*c* the main valve closes—the levers swinging from the nut on the regulating screw. To open up again the lever B is returned to position 7*a*, the main valve opens up and the small tension spring returns the detents, again locking pin B and putting the valve into a position of positive control. This fitting has given general satisfaction and has been extensively adopted.

In future it is intended to make the last type of valve as modified and shown in Fig. 7*d*. A constant closing effort is present, which will allow of perfect regulation under all circumstances and will not necessitate the adoption of saw-formation spigots on the beats. The action under throttling conditions is similar to the former valve, with the difference that the slide valve is returned from position 7*c* to positions 7*b* and 7*a* *by* the spring, assisted by the inlet pressure on the area of the slide-valve spindle where it passes through its stuffing-box.

Hardening-down stresses are alleviated by the expansion of the spindle under temperature. The illustration also shows the relative positions of ports and passages, and consequently gives a better idea of the connection between the positions of the controlling slide valve and the positions of the main valve than is shown in Fig. 7.

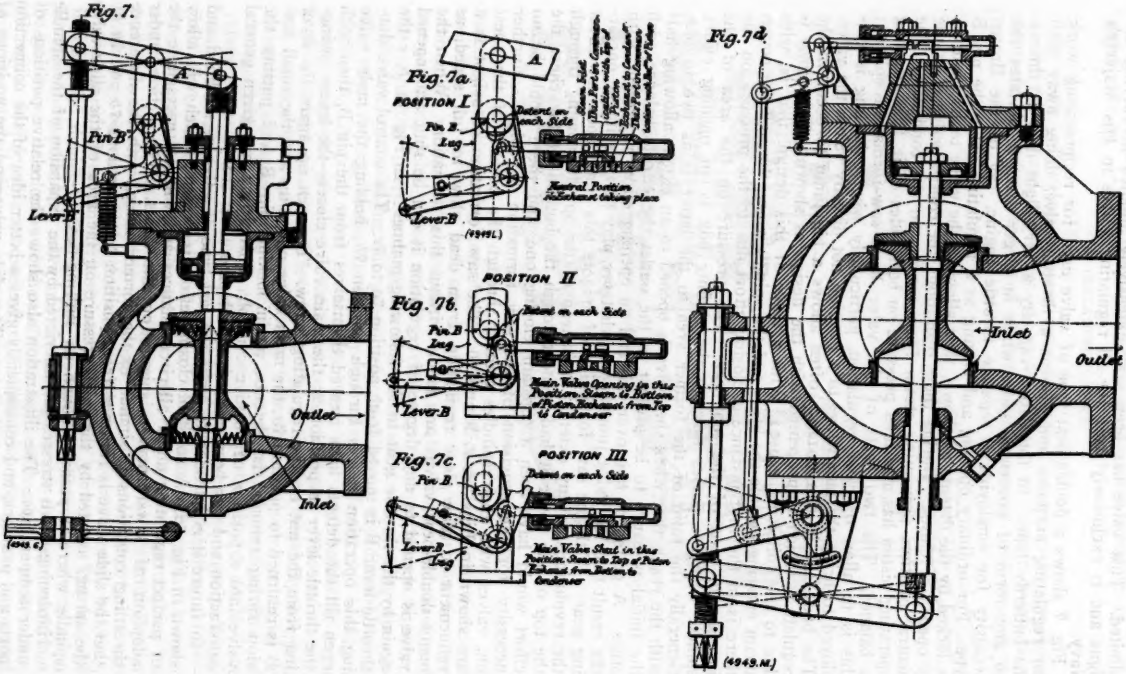


Fig. 7.
A
Pin B
Lever B
Outlet
Inlet
(4945 c)
Fig. 7a.
A
POSITION I.
Pin B.
Leg
Lever B
(4945 L)
Fig. 7b.
POSITION II
Pin B
Leg
Lever B
Fig. 7c.
POSITION III
Pin B.
Lever B
Leg
Fig. 7d.
Inlet
Outlet
(4949. M.)

Fig. 8.

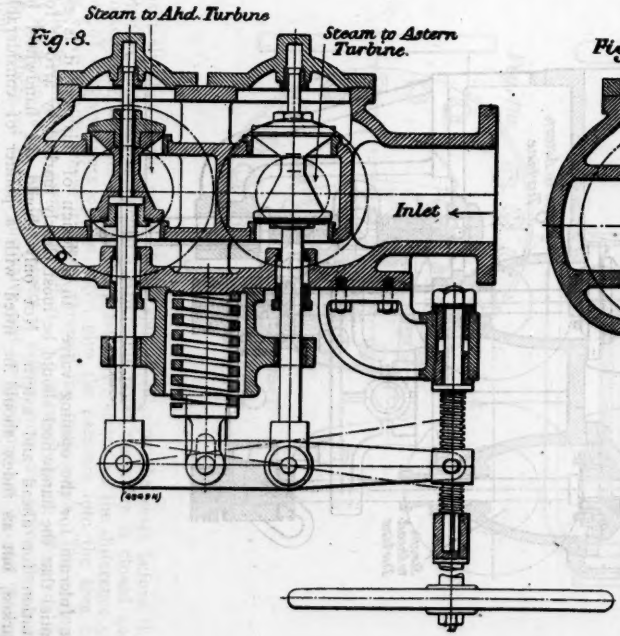
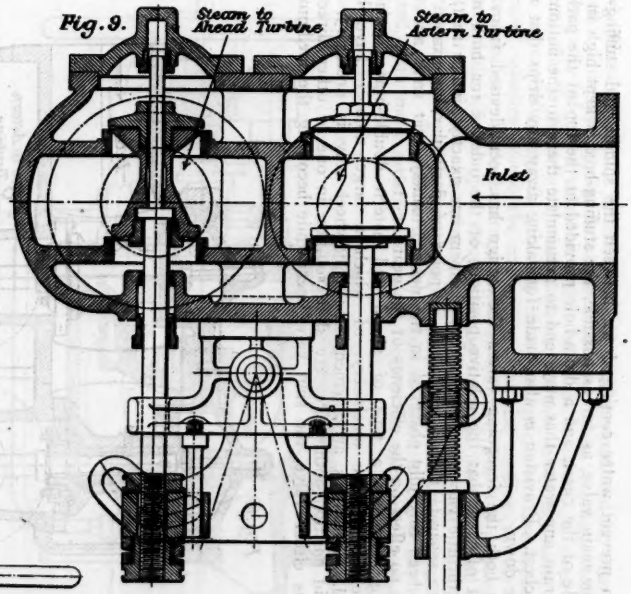

Steam to Ahd. Turbine
Steam to Astern Turbine.
Inlet

Fig. 9.

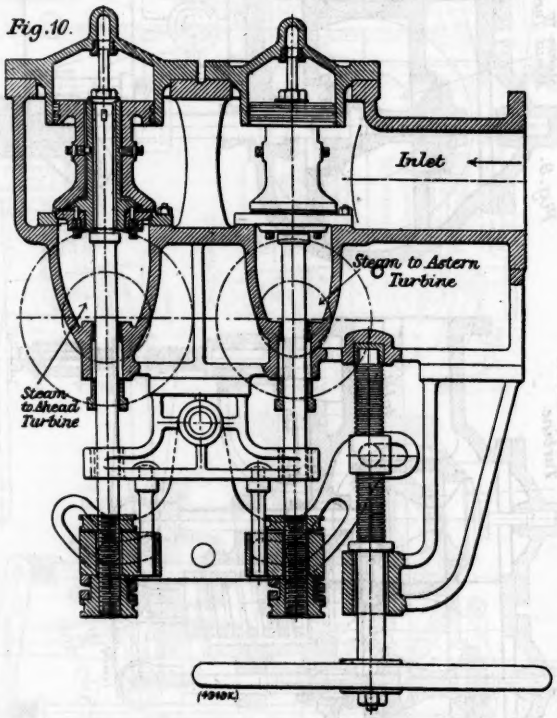

Steam to Ahead Turbine
Steam to Astern Turbine
Inlet

To prevent water coming down between the spindle and stuffing-box of the main valve, as far as possible, the stuffing-box is kept high in the inside of the chest, and a drain-hole provided at the top of the packing to drain any water that may tend to accumulate there to the bottom of the chest. Provision is also made for taking away any drips that may come down.

In both the foregoing valves the design has been devised from the point of view that the relative positions of the indexes for both main and throttle valves at the starting platform are exactly similar to those where separate valves are fitted. A point of particular importance has also been carefully observed in the design, namely, that wear and tear have no effect on the efficiency of the gear.

Fig. 8 shows a type of ahead and astern double-beat regulating valve for turbine machinery, of which a large number have been fitted. With this arrangement one handwheel only is necessary. In a mid position both valves are shut, rotation in one or other direction opens either valve, the shut valve meanwhile becoming the medium of

Fig. 10.



the fulcrum for the opening valve. In a design of this sort it is essential that the handwheel should be most clearly marked by arrows for rotation for ahead and astern. Not only should the handwheel be marked, but an index should be fitted with a pointer of considerable

travel indicating quickly which valve is being operated and in what direction. This also applies to single regulating valves because, while in the majority of cases valves are shut by what is termed a clockwise motion, that is right-hand when looking on the handwheel, with the valve or extending spindle receding from the same, there are cases of extended spindles where the motion is clockwise looking in the opposite direction. With the arrangement as shown in Fig. 8 it is possible, if the spindle of the open valve seizes in the neck bush or gland and it is desired to shut this valve and open the other one, that instead of shutting the open valve first the other valve is opened up—a most undesirable effect. To obviate this the design as shown in Fig. 9 has been adopted, with most satisfactory results. Here the open valve must be

Fig. 11

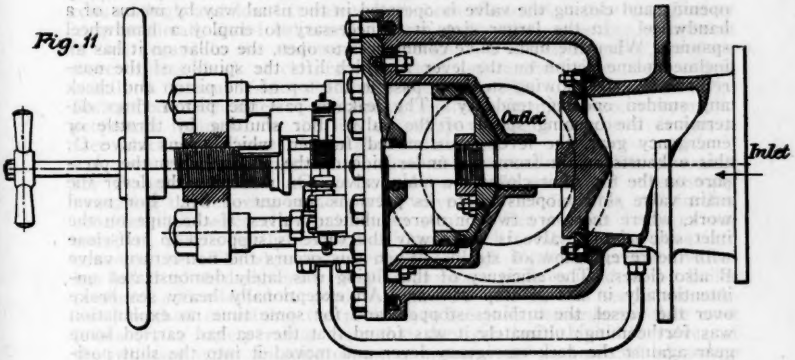
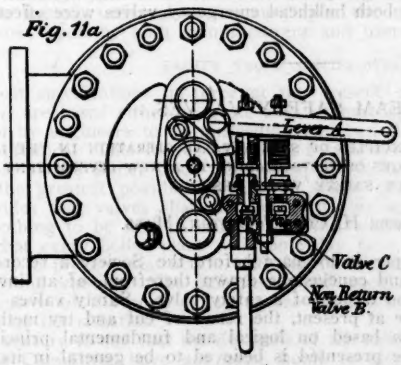


Fig. 11a.



positively shut to the point of the spigots entering the seats before the opposite valve can be opened. After this point the valve is closed and contact made with its seat by the pressure of steam on the difference of area of the two beats of the valve, plus the pressure into the area of the valve spindle where it passes through the stuffing-box. The springs on the spindles also assist this effort. No greater closing effort can be put upon the valve. In this respect it differs from the fitting shown in Fig. 7, where, of course, the valve can be hardened on its seat to any extent.

Another arrangement of balanced valves which has a much greater final closing effort, and consequently ensures tighter valves, is shown in Fig. 10. The valves are balanced by the control valve, which forms part of the spindle, first exhausting steam from the top of the piston. Another outstanding feature is the simplicity of the chest casting, particularly desirable in cast steel. The closing effort when the valves are shut is the pressure on the area of the valves. In these designs of double maneuvering valves and valve as shown in Fig. 7*b* it will be noticed there are no stresses due to expansion of spindles.

A valve widely adopted in conjunction with turbine machinery for both naval and mercantile work is shown in Figs. 11 and 11*a*. This fitting is a combined bulkhead shut-off and emergency valve. It also fulfils the function of a self-closing valve in naval work. For ordinary opening and closing the valve is operated in the usual way by means of a handwheel. In the larger sizes it is necessary to employ a handwheel spanner. When the main valve commences to open, the collar on it has an inclined plane action on the lever A which lifts the spindle of the non-return valve B, allowing steam to pass to the top of the piston and check any sudden opening tendency. The leakage past the piston rings determines the opening speed of the valve. For shutting by throttle or emergency gear, the lever A is moved further, which opens valve C; this exhausts steam from the under side of the piston, when the pressure on the top of it closes the main valve. On returning the lever the main valve slowly opens up to its previous amount of lift. For naval work, where there are two or more bulkhead valves, if the pipe on the inlet side of one valve is shot away the valve is supposed to self-close with the reverse flow of steam. When this occurs the non-return valve B also closes. The efficiency of this fitting was lately demonstrated unintentionally in a fast ship of war. An exceptionally heavy sea broke over the vessel, the turbines stopped, and for some time no explanation was forthcoming; ultimately it was found that the sea had carried some gear against the deck emergency lever and moved it into the shut position, with the result that both bulkhead emergency valves were effectively shut.—"Engineering."

STEAM SAFETY VALVES.

A DISCUSSION OF THE PRINCIPLES OF SAFETY VALVE OPERATION IN THE LIGHT OF AN EXTENSIVE SERIES OF TESTS LEADING TO A NEW DEVELOPMENT IN SAFETY VALVE DESIGN.

By George H. Clark,[1] Boston, Mass.

The object of this paper is to place before the Society a record of the experimental work and conclusions drawn therefrom, of an investigation into the theory and design of a safety valve. Safety valves have been in the past, and are at present, the result of cut and try methods, rather than a mechanism based on logical and fundamental principles. The theory which is here presented is believed to be general in its application, and original in character. In the course of the experimental work undertaken at least 50 valves have been built and tested, and during the course of the experimental work I have not tested, or seen in operation under test, a single valve of a design which is now on the market. Each valve built and tested has been for the purpose of proving or disproving a step in the theory which follows.

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[1]Instructor in mechanical engineering, Massachusetts Institute of Technology.

Presented at the Boston local section of the American Society of Mechanical Engineers, on March 14, 1916.

At a glance, it seems that the problem of the design of a safety valve should be a simple one, but when the facts are brought into line, the problem becomes increasingly difficult. All present day valves employ huddling or expansion chambers, although those best fitted to know the reasons for their use disagree as to the actual part played by such chambers in the action of the valve in which they are incorporated. This is one example of the apparent mysteriousness relative to the action of present day valves. It is hard for the engineering public to understand why it is not possible to increase the lift of valves in accordance with the demands of the day for high capacity valves, without endangering the boilers to which they are connected or the valves themselves, and it is probably true that this misunderstanding is due to the lack of reasonable explanations, in which those best fitted to know concur. The pros and cons of high and low lift valves have been discussed before this Society in great detail, and I believe that I am giving the consensus of opinion in this regard in the statement that high lift valves, so-called, produce more or less shock at closure which may injure the boiler itself, and is usually a serious factor in determining the life of the valve. The Boiler Code Committee of the Society has seen fit, in its recent report, to propose a limitation in the lift of safety valves of approved design, a fact which is evidence of the above statement.

A moment's consideration will show that increasing the lift is, except for changing the seat angle, the only available means of substantially increasing the relieving capacity of a valve of given diameter, and further consideration, when the tendency in steam boiler practice is taken account of, results in the conclusion that high lift valves must come into common use. The absolute necessity for high discharge capacity valves has led English designers to employ auxiliary pistons acted on by boiler pressure, and adapted to produce high lift. To us this seems an awkward method of solving the problem, when it is remembered that most producers of safety valves state that they can build high lift valves of the ordinary type. In spite of the economical and commercial advantage to be gained, such valves are not upon the market and are, in fact, frowned upon by both manufacturers and users.

SAFETY VALVE SPECIFICATIONS.

Most specifications in force at the present day, both abroad and at home, are based either on grate area or heating surface, and are recognized by engineers to be misleading and entirely inadequate for present day practice. This fact is evidenced by the innovations contained in the report of the Boiler Code Committee, which bases the relieving capacity on the greatest possible amount of steam that can be generated and provides that valves shall take care of this maximum amount. There is nothing to be said in criticism of this standpoint, as experience tells us that every boiler plant will some day have to depend on its safety valves to prevent destruction, and that in this emergency they should not be found wanting. The difficulty of this method is the lack of capacity of valves of the present day to provide means for economically taking care of the situation.

I venture to say that the demands of the rules laid down by the Boiler Code Committee have not come home to the boiler users at present. Whatever the position of the boiler user, the basis of the specifications is absolutely right and just, and they distinctly put the real issue on the valve manufacturer himself. If it can be shown by one or a group of valve manufacturers that, after fulfilling all the other requirements incident to a proper safety valve, the lifts and consequent discharge capacities are well above those allowed by the specifications, then the table of capacities must be revised to meet the new conditions in order that true progress may not be blocked.

At this point it is necessary to cite a few examples of the workings

of the proposed rules in order that the true status of safety valve practice may be illustrated. One of the most popular boiler units found in present day plants is the 500 h. p. water tube boiler, and, equipped with stokers, fans, and pressure regulators, this boiler represents current practice. In general this boiler is fitted for and equipped with two 4-inch valves on the drums, and one 4-inch valve on the pipe leading from the superheater. In general such valves are of the 45 degrees seat type, and reference to the table of capacities in the proposed rules allows us to determine the required equipment for this boiler. These boilers are run oftentimes at 300 per cent. rating, and under these conditions generate approximately 45,000 pounds of steam per hour. If the boiler pressure be taken as 200 pounds gage, it will be found that with the maximum lift allowed in the table, the discharge capacity of one 4-inch valve is 9,420 pounds per hour. This boiler will then require five 4-inch valves. Now the specifications state that should two or more valves be connected to a common base, the nozzle supplying the combination must have an area equal to the combined area of the valves connected thereto. Since most of this type of boilers have 4-inch valve nozzles, it will be found somewhat difficult to install the required capacity without extensive changes. Since the limiting size is 4½ inches diameter and since four of this size will be required, there is little to be gained in this direction, as all the connections are unsuitable without the same extensive changes.

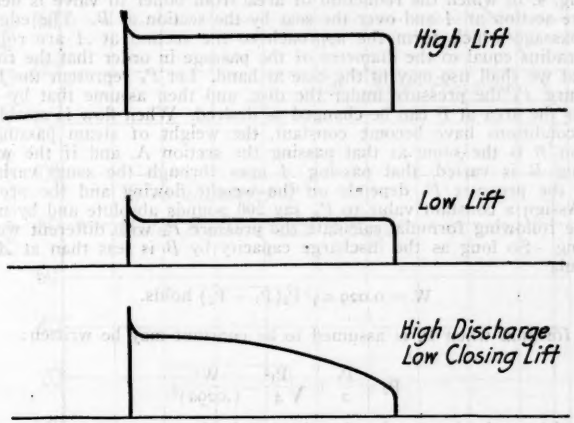
The largest single boiler unit is a Stirling type boiler of 2,356 boiler h. p. adapted to run at 210 per cent. rating. A simple calculation will show that the combined discharge capacity of the safety valves should be 148,000 pounds per hour. This installation under the rules would require eleven valves of the maximum size and maximum lift at 200 pound gage pressure. The safety valves of a boiler of this size would probably be piped outboard, and under the proposed specifications this would require eleven separate discharge pipes. It is customary to set valves connected to the same boiler with a certain range of pressure between their respective popping points. If they are set 2 pounds apart, then the total range of pressure will be 22 pounds or slightly greater than 10 per cent. of the working pressure. The rules require that the installation of valves must relieve the boiler with not greater than 6 per cent. accumulation above the allowed pressure. It is true that this boiler is an extreme at present, but it does represent the tendency of present day practice.

IMPORTANCE OF DISCHARGE CAPACITY.

In the development of boilers and accessory apparatus, the safety valve has become less of a working piece of apparatus and more of a safety appliance. Before the advent of stokers and in the days of hand regulation, either the ash pit doors or the safety valves were open all the time. Today we find ourselves providing a rule which calls upon the engineer to see that his valves are opened once a day at least, to insure that they are in working order. It is clear, then, that the safety valve is called upon in the emergency only, not as a pressure regulator, and if this is the case, should not its discharge capacity be of fundamental importance?

The factors which tend to limit the discharge capacity of the ordinary type of valve when high lifts are attempted are the apparent impossibility of regulation and the shock produced at closure. The actual valuation of the shock produced at closure defies calculation for many reasons, but it can be shown experimentally that the height from which the disc drops to sudden closure plays an important part. The diagrams shown by Figs. 1 and 2 represent graphically the lift characteristics of high and low lift valves, a study of which shows that they are essentially alike in all characteristics, but that in high lift valves the

distance through which the disc drops to sudden closure is much greater than in the low lift valves and actual experience tells us that the low lifts operate without shock, and that the high lifts are subject to that defect in greater or less degree. If this is the case, the logical solution would seem to be a valve giving a diagram like that shown by Fig. 3, in which the lift during the blow-down period gradually decreases to a small value before sudden closure takes place.



FIGS. 1, 2 AND 3.—DIAGRAMS OF LIFT CHARACTERISTICS.

We are not vitally interested in the discharge capacities of valves during the blow-down period provided that high lift can be maintained at the popping pressure. If their discharge capacity is such that, after blowing, the blow-down period is finally reached, the emergency is taken care of, and the installation is safe. In order to produce a valve which will give the diagram shown in Fig. 3, it is necessary to keep the disc in equilibrium with the boiler pressure, while that pressure is being reduced and to explain the factors, it is necessary to go into some detail as regards fundamental theory.

RELATION BETWEEN DISCHARGE AND PRESSURE UNDER DISC.

The first thing to investigate is the variation in pressure on the spring load area of the disc—in other words, the pressure which exists below the disc in advance of and during the period of blowing. By means of certain assumptions we can apply the formula of flow proposed by

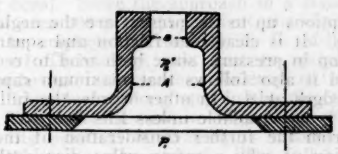


FIG. 4.—REDUCTION IN AREA BETWEEN BOILER AND VALVE.

Napier, and after the calculations are made they may be tempered by the actual conditions which exist, should they deviate from the assumptions.

Steam in the course of its passage from the boiler to the atmosphere encounters two reductions in area of passage, which affect the pressure under the disc, one at entrance to the valve or the nozzle, and the other at the seat. The existent area conditions may be graphically represented by Fig. 4, in which the reduction of area from boiler to valve is denoted by the section at A and over the seat by the section at B. The edges of the passage which form the approach to the section at A are rounded to a radius equal to the diameter of the passage in order that the formula that we shall use may fit the case at hand. Let P_1 represent the boiler pressure, P_2 the pressure under the disc, and then assume that by some means the area at B can be changed as desired. When flow is established and conditions have become constant, the weight of steam passing the section B is the same as that passing the section A, and if the weight passing B is varied, that passing A goes through the same variation. Now the pressure P_2 depends on the weight flowing and the pressure P_1. Assign a constant value to P_1, say 200 pounds absolute and by means of the following formula, calculate the pressure P_2 with different weights flowing. So long as the discharge capacity by B is less than at A, the formula

$$W = 0.029\, a \sqrt{P_2(P_1 - P_2)} \text{ holds.}$$

This formula when P_1 is assumed to be constant may be written:

$$P_2 = \frac{P_1}{2} \pm \sqrt{\frac{P_1^2}{4} - \frac{W^2}{(.029a)^2}}$$

after the quadratic equation has been solved.

Assume now for an actual case, that the diameter at A is 2 inches, giving an area 3.1416 square inches and that P_1 is equal to 200 pounds absolute boiler pressure. Then calculate different values of P_2 with different weights flowing. These values of P_2 may be plotted against the weights and the curve shown by Fig. 5 will result. This curve is interesting since it represents quantities which may be arrived at by calculation only, and which must exist. Its significance is as follows:

With a safety valve of 2 inches nominal diameter and adapted to discharge a certain weight of steam, the pressure under its disc may be found from the curve as follows: Suppose its discharge is 6 pounds per second; then the points C and D are located and the pressure under the disc may be read. It is clear then from this curve that the pressure under the disc varies as the weight of flow varies, and that for valves of low lift the reduction of pressure is not great, but that for high lifts it increases rapidly. This curve is discontinuous when the pressure under the disc reaches a point where it is equal to 0.6 the boiler pressure, under which conditions the area at B has become equal to that at A; the pressure P_2 is the minimum possible, and to all intents and purposes we have a standard orifice, so-called, the characteristics of which are shown by Fig. 6.

Our only assumptions up to the present are the neglect of friction and the rounded edges. It is clear that friction and square edges will tend to increase the drop in pressure, since both tend to reduce the discharge capacity of A, and it also follows that maximum capacity is not available with square edges at A. In other words, the full advantage of the hole in the boiler is not available unless this hole has rounded edges.

There results from the further consideration of the curve in Fig. 5 a simple explanation of why a safety valve disc settles back after the first sudden pop. Since the reduction of pressure on the disc does not

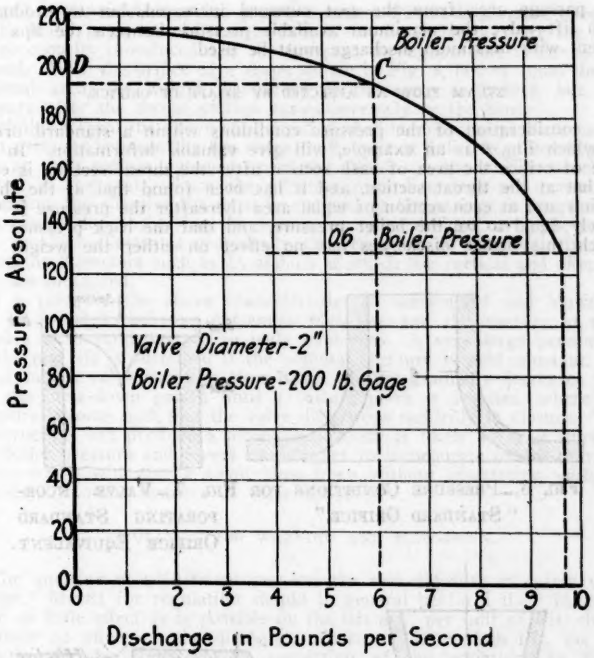


FIG. 5.—RELATION BETWEEN PRESSURE UNDER DISC AND WEIGHT OF STEAM FLOWING.

take place until the valve is open or until flow is established, the lifting force is, at opening, that due to full boiler pressure, and the instant flow is established this is reduced and the disc must necessarily settle back to a position of equilibrium. The design of a valve must then incorporate means to provide lifting force outside the seat area, to make up for the reduction of upward force on the load area due to drop in pressure over and above that necessary to compress the spring.

In any safety valve there are three factors which affect the rate of flow over the seat, which are characteristics of the seat itself. They are: *first*, vertical lift; *second*, seat angle, and *third*, the contour of the passage which approaches the seat. The flat seat is adapted to open the largest area per unit of vertical lift, and hence should be employed, all other things being equal. Since the approach to a standard orifice makes the throat sectional area 100 per cent. efficient, the characteristics of such an orifice should be employed. Professor E. F. Miller, of the Massachusetts Institute of Technology, was first to round the edges of a safety valve in order to increase the efficiency of discharge per unit of lift, and in so doing took a long step in the direction of high safety valve efficiency. The valve shown in Fig. 7, in which the standard orifice, or better, its practical equivalent, is incorporated, follows the theory proposed by him several years ago. The full benefit of the standard orifice would be available, regardless of the type of lip provided, if

the passage area from the seat outward increased, but to produce a high lift valve the maximum available pressure between the lips consistent with maximum discharge must be used.

STEAM FLOW AS AFFECTED BY SHAPE OF ORIFICE.

A consideration of the pressure conditions within a standard orifice, of which Fig. 6 is an example, will give valuable information. In this type of orifice the area of each section after the throat section, is equal to that at the throat section, and it has been found that at the throat section and at each section of equal area thereafter the pressure is very closely equal to 0.6 the boiler pressure, and that the back pressure into which this orifice discharges has no effect on either the weight dis-

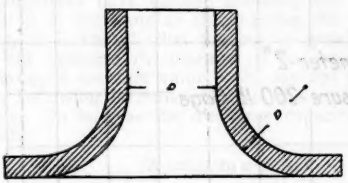


FIG. 6.—PRESSURE CONDITIONS FOR "STANDARD ORIFICE."

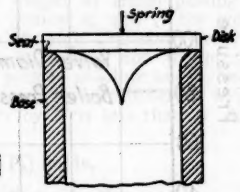


FIG. 7.—VALVE INCORPORATING STANDARD ORIFICE EQUIVALENT.

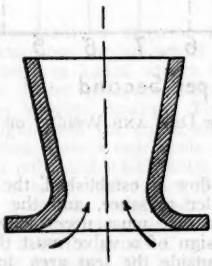


FIG. 8.—PRESSURE VARYING INVERSELY AS AREA.

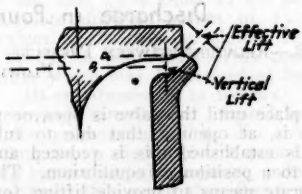


FIG. 9.—VALVE WITH MAXIMUM PRESSURE BETWEEN LIPS.

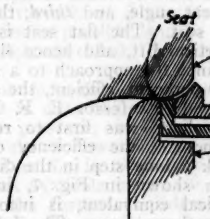


FIG. 10.—VALVE ADJUSTABLE FOR VARIABLE LIFTING FORCE.

charged or the existent orifice pressures unless this back pressure exceeds 0.6 the boiler pressure. If it does exceed 0.6 the boiler pressure the discharge capacity is reduced and the pressure throughout the orifice is increased. With the orifice of a shape shown in Fig. 8, but of equal throat sectional area, the weight discharged is the same as before, but the pressure after the throat section varies inversely as the area.

It follows from the above consideration that in order to produce maximum pressure between the lips there must exist at full lift an equality of area at each section outward from the throat of the orifice which is located at the seat. Since the effective disc diameters are constantly increasing, the effective lift must be reduced as the passage is lengthened. This can be done by giving the passage an upward slope, as is shown by Fig. 9, which illustrates graphically the reduction of effective lift with increasing diameters such as D_1 and D_2 at which the vertical and effective lifts are measured.

If a valve of the above characteristics is constructed and blown, it will be found to have an extremely high lift, and the opening of such a valve is so sudden as to be truly explosive. A very large percentage of the pop lift is sustained if the popping pressure is held constant, and when such a valve starts to blow down the lift gradually decreases during the blow-down period until a certain point is reached, where the pressure become such that the valve disc drops suddenly to closure. This construction will produce a blow-down which is likely to be a third of the boiler pressure and forces its designer to incorporate means of regulation which will give a small blow-down without interfering with the capacity, or other features of valve action.

REGULATION FOR WARNING AND BLOW-DOWN.

The question of adjustment presents the real difficulty in safety valve design. Means for regulation should in general perform their functions with as little effect as is possible on the lift and, per unit of lift, should produce no effect on the discharge capacity. Other than lift, the two functions to be controlled by regulation of the variations in lifting force are warning and blow-down. A valve which chatters or vibrates at the instant of opening, or sizzles at that time shows a lack of lifting force at low lifts, and one which shows an excessive blow-down demonstrates that the lifting force at low lifts is too great to allow of closure, and it follows that the requirements of warning and blow-down are somewhat combative. If it is possible, means should be provided for making, in so far as is possible, the means adopted to regulate one independent of the other. Blow-down is an absolute necessity on the spring loaded type, due to the existence of lifting force outside the spring chamber, and it is clear that attempts to reduce the blow-down will tend to decrease the lift and relieving capacity. In general, the real secret in the design of a perfect safety valve lies, not in the production of excessive lifting force, but in controlling and adjusting it; the action of a valve is dependent, not on the quantity of lifting force, but on variations of lifting force with discharge capacity and pressure.

DESIGN OF NEW VALVE.

Fig. 10 shows a valve which has means for adjustment which are based on the production of variable lifting force as stated above. The two annular rings shown at *A* and *B* carry on their upper surfaces curves which approximate in shape the curve of the disc. The inner ring *A*, which I choose to call the warning ring, is threaded upon the base of the valve just outside the flat seat and carries, threaded upon it, the ring *B*, which is adapted to control the blow-down. These rings may be

lowered away from the disc, or given what I choose to call initial clearance.

Now it is clear that if both of these rings are lowered, an extreme distance, when the valve tends to open no lifting force will be available. If the ring *A* is raised to a position near the valve disc, a lifting force will be provided which will depend, *first*, on the pressure produced between it and the disc, and *second*, on the extent of the disc area on which this pressure acts. As was shown in the discussion of the standard orifice, the pressure depends on the area of the passage and the area in the valve is a function of the initial clearance and the shape of the curve. These factors are built into the valve to produce the desired result, which is to provide lifting force at small lifts to overcome excessive warning, and it has been found that with the proper shape of ring, on its initial clearance, depends the warning of the valve. Since this warning ring alone produces only small lifts, it has no appreciable effect on the blowdown.

Now as the blowdown ring *B* is raised from its lowest position, there is a tendency to build up pressure between it and the disc when the valve is open, which tendency increases as the disc is approached. If a reasonably large clearance is left, it is clear that this ring produces almost no lifting force at low lift, since the area of the passage along this ring is large in comparison to that over the seat, and it follows, also, that as the disc lifts, the effect of this initial clearance on the area of the passage becomes less and less as lifting continues. This can be illustrated by a simple calculation. Suppose that, measured at the extreme outside edge of this ring, the initial clearance is 0.03 inch and that the angle at this point is 45 degrees; then if the disc lifts 0.1 inch in a vertical direction, the height of the passage over the seat will be 0.1 inch, and at the extreme edge the increase in passage height perpendicular to the curves will be 0.07 inch, since with a 45-degree angle the vertical lift is only seven-tenths effective—so that the total height of the passage is 0.1 inch, the same as that over the seat. Suppose now that lifting continues; the area at the edge of this ring gradually approaches that over the seat and can be made to become equal to it by proper design at any chosen lift. It is clear then that for all of the initial clearance of the rings, we have at full lift gotten back to the area conditions of a standard orifice.

Tests of this valve prove the value of this method of regulation in that all the requirements of practice are afforded by it for all of the high lifts. Any one sufficiently interested can lay out a valve of this type, and he will find that if the characteristics of the spring to be used are known, he can calculate with reasonable accuracy the lifting force at any lift and also an approximate value of the total lift. In calculations of this nature, the reduction of pressure on the spring load area, as previously explained in this paper, must be taken into consideration.

DIAGRAMS SHOWING PERFORMANCE OF EXPERIMENTAL VALVES.

I have with some difficulty succeeded in getting some diagrams from valves under service conditions which, I believe, show in detail the results of the methods of regulation which are employed. Fig. 11 shows a photograph of the arrangements of the valves for testing. The tests from which the diagrams were secured were carried on at the power plant connected with the new buildings to be occupied by the Massachusetts Institute of Technology.

One of the regular 4 inch valves was removed and a Y-base substituted in its place; the regular 4-inch was then placed on one of the branches and on the other was placed a special by-passed gate valve, and finally, on top of this, the valve on test was connected. Under the

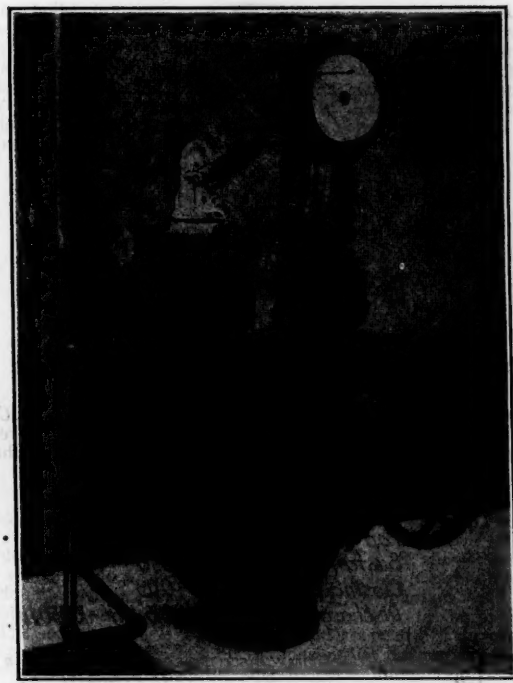


FIG. 11.—VALVE ARRANGED FOR TEST.

above conditions the valve on test was well above 3 feet higher than the top of the drum. Those experienced in safety valve action will realize that this height is apt to seriously affect the action of the ordinary valve. The recording apparatus used consisted of a clock motion adapted to turn a chart once in 15 minutes. Bearing on this chart was a pen rigidly connected to the spindle of the valve and adapted to leave a record thereon of lift with respect to time, the lifts being multiplied by a ratio of approximately four to one.

Fig. 12 shows a chart of the performance of a valve with a 2½-inch seat. This valve was blown at 150 pounds gage boiler pressure and showed, under the existent regulation, 5 pounds blow-down and almost no warning. In order that it may be clearly understood, it has been transferred to a diagram having rectilinear axes, on which is plotted the scale of lifts which resulted from the calibration of the recording lift gage with an accurate micrometer. In Fig. 13 the sudden pop lift is shown at *A*, and the corresponding sustained lift at *B*. The variations in lift at the point *B* are further evidence of the readjustments of pressure which determine the pop and sustained lifts, an idea of which may be gained from the theoretical considerations previously stated in this paper. The discharge capacity of this valve was such that the

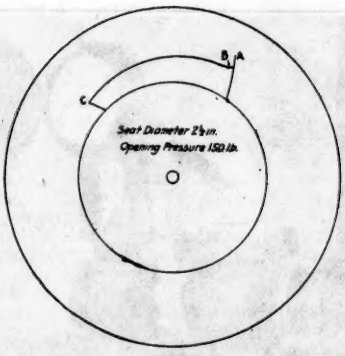


FIG. 12.—PERFORMANCE OF 2½-IN. EXPERIMENTAL VALVE.

popping pressure was not maintained, and the curve from *B* to *C* shows the reduction in lift, which in this type always corresponds to a reduction in boiler pressure. At *C* the valve closed suddenly with less than half its initial sustained lift prevailing.

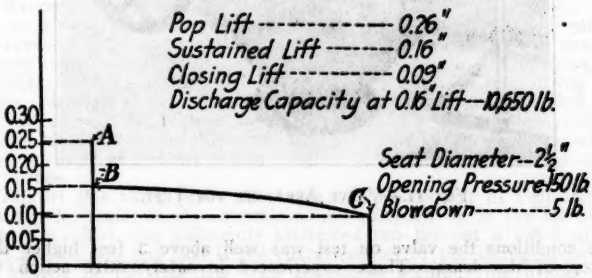


FIG. 13.—ANALYSIS OF DIAGRAM SHOWN IN FIG. 12.

In other words, the drop lift of this valve, which is numerically 0.09-inch, is about equal to that of a low lift valve, while its discharge capacity at popping pressure is nearly twice as great as regards lift, and is even more when seat angle and efficiency are taken into account. This diagram shows the action of the valve when adjusted to give action somewhat similar to that to be found in some types of valves now procurable, and is in a good many ways ideal. Attention is called to the smoothness of the curve between *B* and *C* as evidence of the equilibrium between lift and boiler pressure existent as the boiler pressure is reduced.

Fig. 14 shows precisely the same valve with its outside or blow-down ring lowered slightly. This diagram is, like the one preceding, a photograph of the actual diagram, and has not been transferred to rectilinear axes because of the difficulty in showing the fluctuations in lift. The section of the curve at *A* shows the pop and sustained lifts as before. From *B* to *C* is shown the gradual reduction of lift corresponding to

decreasing boiler pressure, until the point *C* is reached, where the disc suddenly falls from a reasonably high lift to a much lower one. This action is due to the large initial clearance of the blow-down ring, which throws it out of action entirely when the lift value at *C* is reached. The fluctuations in lift at the point *C* show the lack of equilibrium at this point in the action, which results finally in the drop to the lower lift. At the lower lift, represented by *D*, all the supporting force is that due to the inner or warning ring.

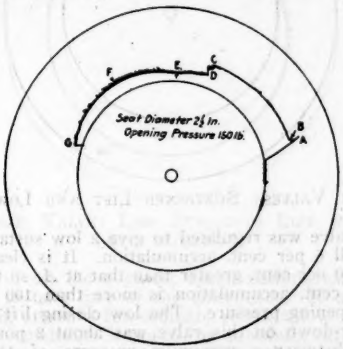


FIG. 14.—VALVE OF FIG. 12 WITH BLOW-DOWN RING LOWERED.

The downward slope of the curve from *D* to *E* shows the tendency of the valve to decrease its lift with decreasing pressure and closure would doubtless have taken place at D, had not the fans been started at this point in an attempt to force the pressure back to the popping pressure. The section from *E* to *F* shows an increase in lift corresponding to the increase in pressure. The fluctuations in lift during this period show the tendency of the blow-down ring to come into action, it being prevented from doing so only by the lack of pressure. The fans were shut off without orders at *F*, and the valve immediately blew down and closed with 0.04-inch closing lift at *G*. If we suppose the diagram to have been completed by closure at *D*, as would have been the case without the attempted accumulation, this diagram would then represent the action of a high lift valve with a closing lift less than that obtainable from any low lift valve of ordinary design. That this action can be and is continually repeated by valves of this type is a fact to which every one who has been a witness of the blowing of this valve will testify. To those who have heard such valves blow no diagrams are necessary.

It was decided after the two preceding diagrams were secured to change to a smaller valve of lower discharge capacity, under which the pressure could be regulated as desired. It is my purpose to bring out by the three diagrams which follow the faculty this type of valve has to respond, under ordinary conditions of regulation, to accumulated pressure above the opening pressure. The new Boiler Code rules, if put into force, will allow 6 per cent. accumulation of pressure, and I wish to show by the diagrams what advantage may be taken of this accumulation.

Fig. 15 shows a chart taken from a 2-inch valve in which the pressure was not allowed to accumulate. The low closing lift in comparison to the sustained lift is to be noted. Fig. 16 shows the same valve with an accumulation of pressure over the opening pressure of 4 per cent. This is under the same regulation conditions as the next preceding chart. The substantial increase in lift and corresponding discharge capacity is to be noted, together with the same closing lift.

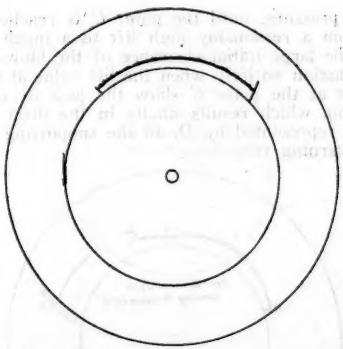


FIG. 15.—2-INCH VALVES: SUSTAINED LIFT AND LOW CLOSING LIFT.

In Fig. 17 the valve was regulated to give a low sustained lift, and was then given the full 6 per cent. accumulation. It is clear that the lift at *B* is more than 100 per cent. greater than that at *A*, so that the discharge capacity at 6 per cent. accumulation is more than 100 per cent. greater than that at the opening pressure. The low closing lift is to be noted as before. The blow-down on this valve was about 2 pounds, so that this chart was made between a maximum pressure of 159 pounds and a minimum pressure of 148 pounds. The new Boiler Code specifications rate valves on their sustained lift at the popping pressure, and so it is clear that with this type the basis is amply safe.

In presenting the diagram (Fig. 18) for consideration I believe that I am showing a safety valve which is absolutely new, both in conception and results. It has been the aim of safety valve designers to produce a valve which relieves a boiler with an action which is smooth and containing as few sudden fluctuations in pressure as is possible. The Boiler Code Committee recognized the capabilities of standard valves to respond to accumulated pressure by incorporating the 6 per cent. accumulation rule. The ideal safety valve for practical purposes is, I believe, one in which a reasonably low lift is maintained at the popping pressure,

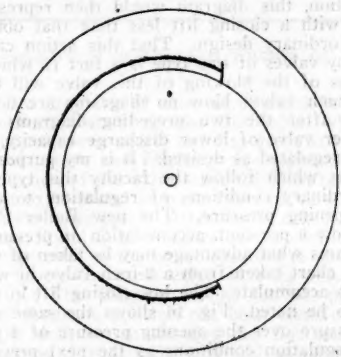


FIG. 16.—2-INCH VALVE: PRESSURE ACCUMULATION BETWEEN OPENING AND CLOSING.

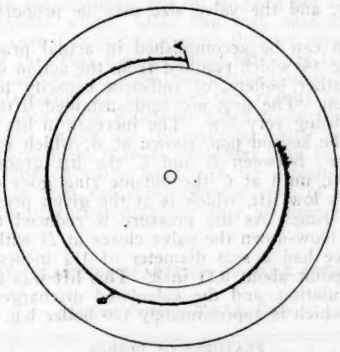


FIG. 17.—2-INCH VALVE: LOW SUSTAINED LIFT WITH PRESSURE ACCUMULATION.

but one which will on the slightest accumulation respond with an extremely high lift to take care of the emergency. If a low lift is available at the popping pressure for ordinary relief, a low blow-down may be incorporated, the condition imposed by absolute safety being that at slightly above the popping pressure, the lift will have increased to such an extent that any possible emergency will be taken care of.

Fig. 18 shows a diagram from a valve which produces the above action, and is what may be called a second opening valve. Many of this type have been built, and it has been found that with the proper regulation of rings, all valves having two rings will produce this action in modified form. Imagine, first, one of the ordinary type blown with the outside ring removed. This would result in a low lift and a very low blow-down. Now suppose that the blow-down ring is in place, but that it is given such a large initial clearance that the valve will act, provided that it has sufficient capacity to relieve the boiler, exactly as before, but if it has not the required capacity the pressure will accumulate and the increase of lift resulting with only the inner ring in action will finally bring the outer ring into action, and a second pop will occur. This pop will result in a high lift, with corresponding high

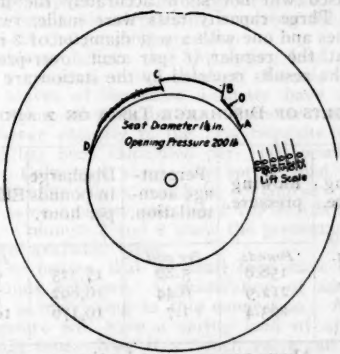


FIG. 18.—A "SECOND OPENING" VALVE.

discharge capacity, and the valve size may be proportioned to give the required safety.

That this action can be accomplished in actual practice is shown by the diagram in Fig. 18, which resulted from the action of a 1¼ inch valve on a boiler, or rather boilers, of sufficient capacity to produce the required accumulation. The first pop and sustained lifts are shown at *A*, the sustained lift being very low. The increase in lift between *A* and *B* finally results in the second pop, shown at *B*, which is the result of accumulated pressure. Between *B* and *C* the lift gradually falls as the pressure is reduced, until at *C* the outside ring goes out of action and the disc drops to a low lift, which is at the given pressure just sustainable by the inner ring. As the pressure is reduced the lift decreases, and after a slight blow-down the valve closes at *D* with a very low closing lift. This valve had a seat diameter of 1¼ inches and lifted at its highest sustained point about 0.11 inch. This lift was the result of about 4 per cent. accumulation, and the calculated discharge capacity is 5,100 pounds per hour, which is approximately 170 boiler h.p.

FEATURES OF DESIGN.

Careful design will allow of bringing the second pop down very near the first one, and the limits of regulation may be so built into the valve by limiting the possible up and down motion of rings, so that this second pop will always come within a given range of pressure. This type of action has many advantages to recommend it. The absence of sudden excessive changes in relieving capacity and the low lift at closure are evident from the diagram. It is clear that if a boiler had two valves of this type, the second might be set to blow at a pressure corresponding to the point *O* on the diagram, so that the two valves would be able to handle the boiler at all ordinary times without the second pop, but in the case of emergency both would rise to the second pop lift. The installation would then be essentially a low lift installation, but would have all the advantages of high lift in the case of emergency.

DISCHARGE TESTS.

In the course of my experimental work very few discharge tests could be run, since most available plants have no means of measuring the water used. Three tests were run at Annapolis, Md., at the United States Naval Experiment Station, the results of which are useful in this paper only in so far as they establish the fact that the efficiency of the standard orifice type is so nearly 100 per cent. per unit of lift that the accepted formula used will not show accurately the deviation from the above efficiency. Three capacity tests were made, two with a seat diameter of 2½-inches and one with a seat diameter of 2-inches. Two of the tests were run at the regular, 6 per cent. over-pressure common to Navy practice. The results reported by the station are shown in Table 1.

TABLE I.—RESULTS OF DISCHARGE TESTS ON 2 AND 2½-IN. VALVES.

Valve diameter.	Opening pressure.	Blowing pressure.	Percentage accumulation.	Discharge in pounds per hour.	Efficiency.	Lift.
Inches.	*Pounds.*	*Pounds.*	*Per cent.*		*Per cent.*	
2.5	150	158.8	5.86	17,325	99.43	.252
2.5	200	212.9	6.44	16,892	78.88	.238
2.0	200	203.4	1.7	10,176	101.57	.145

Above pressures are by the gage.

The test in which the efficiency fell off to 78.88 per cent. is reported by them to be a mistake which, after the test, was impossible of correction, so I think that it is fair to assume that we may accept the results as evidence of the fact that the discharge efficiency is so nearly 100 per cent. that for ordinary calculations the efficiency may be neglected.

It is always interesting to compare the results obtained from any piece of apparatus with those which we know to be theoretically possible. In the case of safety valves the full opening valve serves as a convenient basis of comparison, in that its capacities may be calculated without reference to the type of valve used. If we attach a standard orifice to a boiler we have then the most efficient way of discharging steam, and this on a basis of throat diameter may be characterized as the full opening valve.

TABLE 2 CALCULATED CAPACITIES OF FULL OPENING VALVES.

Opening Pressure 150 Lb. Gage

Nominal Valve Diameter	1"	1½"	2"	2½"	3"	3½"	4"
1 Discharge capacity of a full opening valve	6,660	14,900	26,600	41,600	59,900	81,600	106,500
2 Discharge capacity of A.S.M.E. valves (from table)	904	1,625	2,529	3,613	5,419	6,954	8,670
3 Discharge capacity of proposed type (from tests)	...	...	5,310	10,640	...	...	...
4 Spring load of (1) in pounds per 1000 lb. discharged	17.7	17.7	17.7	17.7	17.7	17.7	17.7
5 Same for A.S.M.E.	131	120	185	204	196	208	218
6 Same for proposed type	...	...	88.5	69.2	...	...	...
7 Percentage of full opening attained by proposed type	...	...	26.6	25.6	...	...	...
8 Same for A.S.M.E.	13.6	10.9	9.51	8.67	9.05	8.50	8.15

In column 1, Table 2, are shown the capacities of full opening valves of several diameters figured directly from Napier's formula. Column 2 shows the discharge capacity of valves of the same diameters as read from the table in the recent A.S.M.E. Boiler Code. I have assumed that these values represent standard conservative practice. Column 3 shows the discharge capacity of the type of valve which I have described, and is the result of calculations involving the lift as measured from the diagrams at the popping pressure.

It seems reasonable to believe that the spring load of a safety valve is to some extent a measure of the shock at closure, which may be expected. Since all valves of the same diameter have approximately the same spring load, some other basis than diameter must be chosen for comparison. Discharge capacity is the first requisite in all valves, and so the spring load has been calculated per 1,000 pounds of steam discharged per hour. Column 4 shows the spring load per 1,000 pounds discharge of full opening valves, which is, of course, constant. Columns 5 and 6 show the same quantity calculated for A.S.M.E. valves and for the proposed type. Columns 7 and 8 show the percentage of full opening attained by the two available types.

There is reason to believe that we shall soon have boilers which will operate at 600 pounds pressure. Consideration of some of the values given in this table is interesting in this connection. A 4-inch valve set at 600 pounds pressure will have a spring load of approximately 7,540 pounds, or about 3½ tons. Now if a 2-inch valve can be constructed to

give twice the lift of the 4-inch, it will have the same discharge capacity, but its spring load will be only one quarter as great, or 1,885 pounds. In other words, the discharge capacity varies as the first power of the diameter and the lift, but the spring load, which is independent of the lift, increases as the square of the diameter. It seems clear from this that high pressure boilers will of necessity require small-diameter high-lift valves.

This table shows the net result of the development in spring loaded safety valves, and serves to emphasize the opportunities which may be grasped by the designer.

The question of the effect of back pressure in the casing of a safety valve has been much discussed. It has been said that it may operate in two ways to reduce the discharge capacity. *First,* the effective lifts may be reduced, and, *second,* it has been held that back pressure reduces the efficiency of discharge of the orifice. If any of the disc area is exposed to back pressure, the lifts will be reduced to some extent, but consideration of the theory of flow through an orifice, and particularly a standard orifice, brings out the fact that as long as the back pressure in the casing is less than 0.6 the boiler pressure, the discharge of the orifice is not affected.

The discharge capacity tests made some years ago by Mr. P. G. Darling went some distance in establishing this fact, although no general conclusions were drawn. It remained for a section of a Committee of the American Society of Refrigerating Engineers, with Professor E. F. Miller as chairman, to establish this fact beyond question. In refrigeration installations long discharge pipes are required, and the back pressure built up in them as a result of pipe friction produces so much back pressure that either the ordinary type of valve is useless or its discharge pipe must be of an excessive size. The experimental work resulted in a valve which was so constructed as to take advantage of back pressure to increase its lift without in any way having its discharge capacity per unit of lift reduced. This valve is due to the ingenuity of Mr. F. L. Fairbanks of the Quincy Market Cold Storage and Warehouse Co., and has been adopted as the standard in Massachusetts.

EFFECT OF BACK PRESSURE.

The possibility of back pressure operates to reduce the discharge pipes to reasonable sizes in spite of their length. There are many economical advantages to be gained for steam users by a valve which will operate without harmful effect of back pressure. *First,* the sizes of discharge pipes may be reduced, and, *second,* several valves may be connected into a common discharge pipe.

EXAMPLES OF THE NEW VALVES.

Fig. 19 shows a complete view of a valve which has the above advantages. In the first place, valves of high lift, and by that I mean higher lift than any of those obtainable at present, must be guided from above, as guides in the base will cause too great a drop in pressure on the disc area. In this view the disc is made in the shape of a hollow piston, which is guided by a cylindrical sleeve which encloses the spring chamber. The fit existent between these is extremely loose, and no attempt is made to prevent leakage by the disc. This loose fit insures that the disc will not bind because of expansion or contraction between the members due to temperature changes, and is long enough in relation to the diameter of the disc so that it will never cock. Its looseness alone insures that the disc will come to a uniform bearing on the seat, and it has been found that under no conditions of back pressure has the leakage by it been sufficient to produce pressure in the spring chamber, which

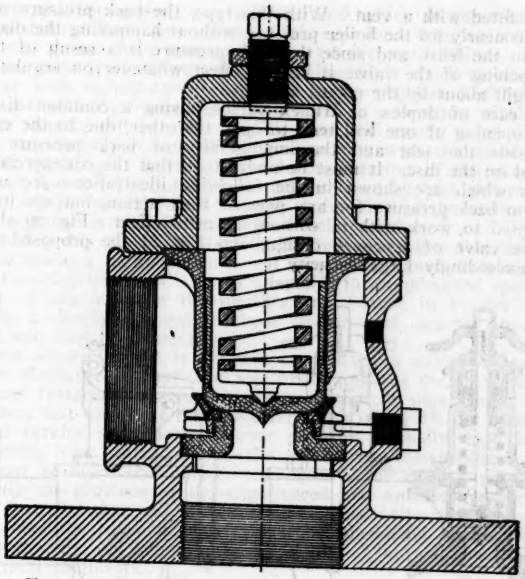


FIG. 19.—STEAM SAFETY VALVE OF APPROVED DESIGN.

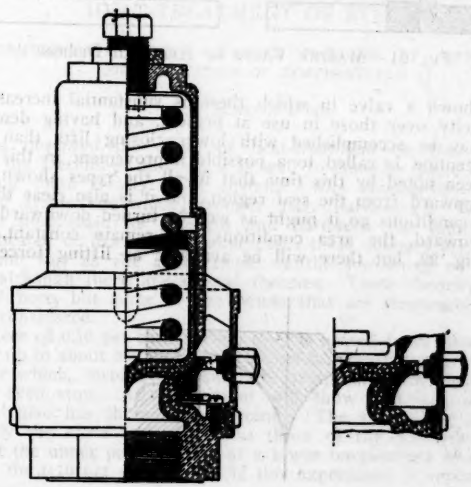


FIG. 20.—LOCOMOTIVE VALVE OF APPROVED DESIGN.

is always fitted with a vent. With this type, the back pressure may be built up to nearly 0.6 the boiler pressure without hampering the discharge capacity in the least, and since the back pressure is a result of the effective opening of the valve, it has no effect whatever on regulation, if it is brought about by the means shown.

In the case of duplex or triplex valves having a common discharge pipe, the opening of one will tend to open the other, due to the exposed area outside the seat and the impossibility of back pressure acting downward on the disc. It must be understood that the commercial types of valves which are shown in the following illustrations are not dependent on back pressure for any part of their action, but are in every case adapted to work with it, without harmful effect. Fig. 20 shows a locomotive valve of approved design, and Fig. 21 the proposed marine type of exceedingly large capacity.

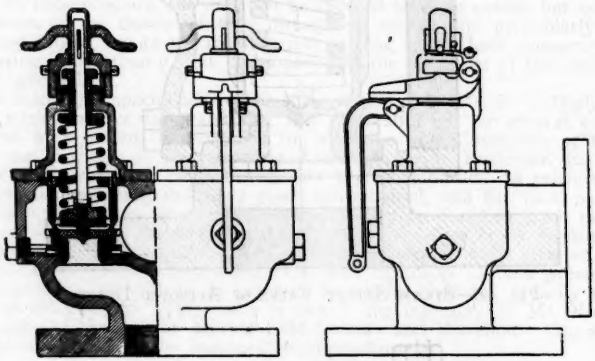


FIG. 21.—MARINE VALVE OF APPROVED DESIGN.

Having shown a valve in which there is substantial increase in discharge capacity over those in use at present, and having demonstrated that this may be accomplished with lower closing lifts than are now the rule, attention is called to a possible improvement in this type. It will have been noted by this time that in all the types shown, the passage turns upward from the seat region, and it is also clear that so far as the area conditions go it might as well be turned downward. If it is turned downward, the area conditions will remain constant, as illustrated in Fig. 22, but there will be available as lifting force a factor

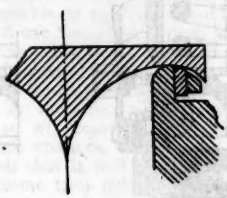


FIG. 22.—VALVE WITH "JET ACTION."

which would act upward on the disc, and which is commonly known as a jet action. Some few valves of this type have been constructed, and the results gained have been convincing of their possibilities. Enormous lifts have been attained with the same absence of shock at closure as is the case with valves herein described. The problem has been by no means solved, but advances have been made in a direction which tends to indicate that it will be solved finally, and will result in enormous discharge capacities per inch of valve diameter, with the same efficiency per unit of lift. It is clear from a consideration of the diagram that the force due to jet action will not be destroyed by back pressure, as the velocity of the issuing jet of steam is a function of the orifice pressure, which pressure is not affected by the back pressure.

In closing this paper, attention is directed to the fact that this valve is not essentially a high lift valve. It may be designed to give low lifts as well as high lifts, and is easily adaptable to the proposed specifications. It will, of course, under the specifications, benefit, by reason of its flat seat, by a discharge capacity about 30 per cent. greater than the 45-degree seat now in common use.

I have shown that it is possible to produce a valve of high lift as regards discharge capacity and low lift as regards the shock at closure. This one feature alone removes the most serious objection to high lift. It is clear that for all of the reasonably high initial lift the valves have a marked faculty to respond to over pressure. Finally, high lift valves must come into general use as the safety requirements in boiler practice must be more economically met than is the case at present. It is distinctly the province of the experienced and unbiased engineer to lay down the rules that will govern future safety valve installations, but in determining the rules the valves must be considered on a basis of discharge capacity and action rather than lift.—"Journal American Society Mechanical Engineers."

HEAT-TREATMENT OF STEEL.*

PHOTOMICROGRAPH STUDY—CHANGES PRODUCED BY ANNEALING—DETERMINATION OF TEMPERATURES.

By Martin Syte.

Having observed the effect of heat-treatment on the properties of steel, in the December number, it is well to consider some of the causes underlying these results. For instance, why should the steel become hard and brittle and much stronger after hardening? Why should a hardened piece lose its strength and hardness, but gain in ductility, when reheated or drawn? In some cases these changes are very difficult to account for, as the exact cause of the hardening of steel is not known, although there are several theories. These theories will not be discussed here, but some of the points that are thoroughly understood will be considered.

If a piece of 0.10 per cent. carbon steel is heated from atmospheric temperature up to about 930 degrees C. (1,706 degrees F.), there will be three points at which, instead of rising, the temperature of the piece pauses or may even stop. An experiment will show that 0.25 per cent. carbon steel also has three critical points. The two lower points are at practically the same temperature as those of the 0.10 per cent. carbon steel, but the upper point will be at a lower temperature than the highest point of the 0.10 per cent. steel. If this experiment is repeated with 0.50

*The first installment of this article appeared in the December, 1916, number of "Machinery."

per cent. carbon steel, only two points will be found. The first will be at practically the same temperature as that of the 0.10 and 0.25 per cent. carbon steels, but the second point will be much lower than their second point. In the case of 0.85 and 1.50 per cent. carbon steels, just one point will be found, but this will coincide with the first point of the lower carbon steels.

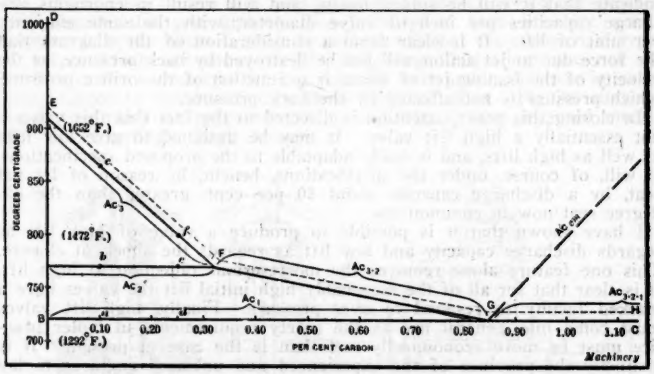


FIG. 1.—POSITION OF CRITICAL POINTS IN HEATING STRAIGHT CARBON STEEL.

Fig. 1 shows the location of these points graphically after a diagram by Sauveur. This diagram should be firmly fixed in mind, as it shows the relative location of the critical points of the plain carbon steels. The carbon content of the steel is shown, in per cent. along the lower horizontal line *BC* and the temperatures at which the critical point occurs along the vertical line *BD*. For instance, the points *a*, *b* and *c* are the three critical points for the 0.10 per cent. carbon steel and *d*, *e* and *f* for the 0.25 per cent. carbon steel. The diagram was obtained by getting the critical points of many carbon steels and then joining these points by lines.

The question naturally arises as to how these points are obtained. For instance, how is it possible to locate the three critical points of the 0.10 per cent. carbon steel? There are several methods for doing this, and the ones most commonly employed in practice make use of the thermo-electric pyrometer. As it is practically impossible to use a furnace that will heat the piece at a uniform rate, it is apparent that there may be pauses or retardations in the heating or cooling that are not caused by any properties of the steel under test. In order to eliminate such pauses from the test, the temperature of the steel should be compared with the temperature of a body that has no transformation points within the temperature range covered by the experiment. If this so-called neutral body is put in the same furnace as the test piece, both bodies will be about equally affected by any irregularities in the heating of the furnace. In other words, they will be heated together, and when the temperature of one varies, the temperature of the other will vary also, so that there will never be a very marked difference in their temperatures; but when a critical point of the steel is reached, there will suddenly be a great difference in their temperatures, as the neutral body will be unaffected (possessing no critical points), while the steel will cease to rise in temperature. By using a double pyrometer to measure the difference between the two bodies, it is always possible to detect when

a critical point has been reached; and a second pyrometer will show at just what temperature this point occurs. By this means the critical points of any steel can be found.

The critical points that occur in the heating of a steel also occur in the cooling, but at a lower temperature. If the point occurs in heating, it is called an *Ac* point; and in cooling, an *Ar* point. The point Ar_1 that occurs in cooling is called the "recalescence point," due to the fact that the steel actually gives off heat, or recalesces, at this point. Many writers use the term recalescence point very loosely, but it should be borne in mind that the only correct recalescence point is the Ar_1 point. The importance of knowing the location of these critical points lies in the fact that the correct hardening temperature is just above the upper critical point for any steel; or along the dotted line *EFGH*. Theoretically, it is just at the upper critical point, but it is safer to go a little higher.

CHANGES THAT OCCUR AT CRITICAL TEMPERATURES.

In studying the changes that take place at the critical temperatures, the range of temperature from the lowest critical point to the highest point of any steel may be called its "critical range." If any piece of steel is heated just below the foot of the range, that is, Ac_1 or $Ac_{1\text{-}2\text{-}3}$, and quenched in water, it will be found that there are practically no changes in its properties. If the steel is heated to this point or a little above and then quenched, it will be found that the piece has been made stronger, harder and less ductile; that is, its tensile strength, elastic limit, and hardness have been increased while its elongation and reduction of area have been decreased. If the steel is hardened at successively higher temperatures, it will be found that it is made stronger and harder but less ductile, in each case, until the top of the critical range is reached, when the maximum degree of strength and hardness and the minimum of ductility are obtained. If the steel is heated considerably above the top of the range, there is no gain in hardness or strength; the only effect is that the grain is made coarser. In other words, steel can be hardened somewhat by heating it above its first critical point and quenching, but the full hardness or strength is not obtained until the steel is heated above the upper critical point. This fact is true of all straight carbon steels.

To determine the reason for this fact, it is necessary to study the steel in the normal state. Normalized steel consists of pearlite and ferrite in the case of hypo-eutectoid steel, pearlite alone in the case of eutectoid, and pearlite and cementite in the case of hyper-eutectoid. When a piece of any of these types of steel is heated to a temperature below the first critical point, no change takes place in the microstructure, and hence there is no change in the physical properties. But if it is heated to the point Ac_1, there is a very marked change in the properties. This is due to the marked change in the microstructure, for just as soon as the point Ac_1 is passed, the pearlite, instead of remaining as a mechanical mixture of alternate layers of cementite and ferrite, becomes a solid solution. As this change of the pearlite has increased the strength of the metal, it follows that the solid solution of the pearlite constituents must be stronger than pearlite in the normalized state.

As the name implies, a solid solution has all the characteristics of a liquid as regards the intimate mingling of its constituents. In fact, in a solid solution it is impossible to distinguish, with the highest power of the microscope, one constituent from another. Hence, while before the pearlite became a solid solution, the plates of carbide (cementite) could be distinguished from those of ferrite, after it was converted into a solid solution neither could be distinguished, because they were so intimately mingled. In fact, in a solid solution each crystal contains a portion of each ingredient, and it is impossible, by any physical means, to separate the matter contained in the crystal.

As the temperature of the steel is raised above the foot of the critical range, some of the free ferrite begins to enter this solid solution and more and more of it enters the solution as the temperature is increased. In fact, when the top of the critical range is reached, all the free ferrite has entered the solid solution, and if the steel is then quenched, it will be in the strongest possible state. If the steel contained no free ferrite or free cementite, that is, if it were a eutectoid steel, all the pearlite would go into solution as soon as the foot of the range was passed (just as it did in the hypo-eutectoid steel), and the whole steel would then be in the state of a solid solution. If it were a hyper-eutectoid steel, the pearlite would go into solid solution as soon as the foot of the range, Ac_{3-2-1}, was crossed, but the excess cementite would not go into solid solution until the dotted line Ac_{cm} was reached.

To summarize, no matter whether the steel is hypo-eutectoid, eutectoid or hyper-eutectoid, when heated the pearlite goes into solid solution at the foot of the range, Ac_1 or Ac_{3-2-1}. If there is any free constituent, as ferrite or cementite, it will be gradually absorbed into the solid solution as the steel is heated to the top of the range (Ac_3 or Ac_{3-2-1} in the case of hypo-eutectoid steel; Ac_{3-2-1}, which is both top and bottom of range, for eutectoid steel; and Ac_{cm} for hyper-eutectoid steel), and by the time the top of this range is reached, all the pearlite will be absorbed. In other words, no matter what kind of steel we have, it will be in the state of a solid solution as soon as it is heated to the top of its critical range.

If the steel is quenched or cooled with extreme rapidity, it will be retained, or locked, in this state, so that when it is cold it is still a solid solution; it is then called austenite. But in practice it is impossible, as a rule, to quench or cool the steel quickly enough to retain it in this perfect solid-solution state, and some of the constituents separate out of this solid solution. If the steel is cooled in liquid air, which is very cold, perfect austenite may be obtained in some cases. If the steel is cooled in water, such a close approximation to the solid solution cannot be obtained, and if cooled in kerosene, the approximation will be found to be even more imperfect. With heavy oil or air cooling, a still more imperfect solid solution is obtained.

By looking at a specimen of hardened steel under the microscope, it is possible to determine just how close it is to the solid solution. The perfect solid solution is called austenite, the first stage in its decomposition is martensite, then come troostite, sorbite, and pearlite. Each of these states has its characteristic appearance under the microscope.

It has been shown that by heating steel to the top of its critical range the steel is put in the solid-solution state, and by quenching in mediums of varying effectiveness the steel may be obtained in the cold state as a solid solution, or one of its decomposition products. But it is possible to obtain the exact decomposition product desired by heating to the top of the range, cooling rapidly enough to get austenite or martensite, and then reheating below the range so as to get troostite or sorbite. In other words, it is possible to obtain the steel in the exact state desired by simply hardening it with the proper rapidity, or by hardening and then reheating or drawing the steel. The hardening process converts the steel into a fine-grained solid solution (or to an approximation), and the drawing treatment lets the steel decompose into one of the states that experience has shown is best suited for the work desired.

In the specimens used for determining the chart shown in Fig. 1, the steel was first hardened just above the top of the critical range, which was determined by means of a neutral body and pyrometers. After this steel was hardened it was as close to a solid solution as it was desirable to get this particular steel commercially; it was then in the state of martensite. Some pieces were then drawn at each of the temperatures shown on the chart, and the martensite was decomposed by this

drawing into the softer and weaker, but more ductile, decomposition products, troostite and sorbite. It should be mentioned that each of the successive decomposition products of austenite is softer than those preceding it, except martensite, which is harder than austenite.

CHANGES PRODUCED BY ANNEALING PROCESS.

In a good many cases where steel is to be annealed, the only object desired is to soften the steel so that it can be machined or worked more easily. In other cases, it is desirable not only to make the steel softer, but also to refine the grain size and to make the steel soft. In order to anneal steel perfectly for the latter purpose, it should be heated to the top of the critical range, held there until every part has been thoroughly heated to this temperature, and then very slowly cooled, preferably in the furnace. This will not only wipe out any coarse structure and make the steel fine grained, but it will leave the steel as soft as possible. This is due to the fact that when the steel is heated to the top of the critical range, it goes into the state of a fine-grained solid solution. Then by slow cooling the steel is given no chance to set in this solid-solution state, but forms soft, ductile sorbite or pearlite. If the steel is cooled more rapidly, it will not be given sufficient time to decompose from the solid-solution state, and troostite or sorbite, or a combination, is obtained. If the steel is cooled in the air, it will be still harder, due to the fact that cooling in air is a sort of mild form of hardening. This is shown very clearly in Fig. 1, where the steel has been hardened, then reheated to the top of the range and air cooled. It will be noted that both the tensile strength and the hardness are increased.

The most perfect way to anneal is to heat the steel to the top of its range and harden; this gives a fine-grain hard steel. It should then be reheated to just below the foot of the range and cooled very slowly; this gives the finest grain possible, together with extreme softness, and is a method much used in France. It has been found best to harden hyper-eutectoid steel of 0.85 per cent. carbon or more just above the line Ac_{3-2-1}, for if the top of the range Ac_{cm} is passed, all the excess cementite may be in solid solution, but this higher heat will enlarge the grain so much that the advantage gained does not pay. The one exception to this is the case of high-speed steels, which, due to their special ingredients, may be heated above Ac_{cm} and all excess elements changed into solid solution, without dangerously enlarging the grain size.

The literature on hardening is sometimes very inexact and unclear, but if it is remembered that the line *EFGH* shows the proper point for hardening, no confusion should result. In all cases the hardening or correct annealing temperature is the top of the range Ac_3 or Ac_{3-2} in steel under eutectoid carbon content, and Ac_{3-2-1} on all eutectoid and hyper-eutectoid steels with the exception of high-speed tool steels.

A TYPICAL HEATING TROUBLE.

In considering the practical operations of hardening, it will be found that the theoretical knowledge gained will stand us in good stead. We may harden a brand of steel for months with complete success and then suddenly have trouble at some inopportune time. The steel may not harden the way it did before, or it may crack in the hardening. The exact cause is sometimes difficult to find, and the man whose practical knowledge is not coupled with an understanding of the why and wherefore of his hardening operations frequently finds himself in trouble.

For instance, a hardener had been getting entirely satisfactory results by hardening arbors in oil. He had hardened thousands of them without experiencing any trouble when suddenly they tested soft after the heat-treatment. The hardener was the center of all the resulting com-

plaints, and naturally he claimed that the analysis of the steel was different from what he had been getting. By using water instead of oil and heating considerably higher, he managed to make a fairly satisfactory job, but still a considerable number cracked in the hardening. The steel manufacturer insisted that the steel was within specifications, and actually showed by analysis that the former pieces that hardened in oil and those which required water quenching were the same steel. The hardener, who merely did this work by rule, was naturally at a loss to overcome this trouble and eventually lost his position. Investigation later showed that the steel in both cases was identical, but that the supplier, having run short of the size bar generally supplied, had obtained permission to supply a smaller one. As very little stock was turned off from this, the decarburized outer skin, or bark, was not completely removed. Therefore, instead of the steel being 0.65 per cent. carbon, as specified, the exterior was 0.35 per cent. carbon and the interior was 0.65 per cent. It therefore required water and a higher heat to make this low-carbon outside portion hard enough.

If the hardener had only studied the theory a little, he would have known that this higher heat and water quenching was an indication that the carbon was lower. His limited general knowledge kept him from knowing that his first contention that the analysis varied was correct, as far as the outer surface of the bar was concerned, and that the steel manufacturer was also correct as far as the analysis of the interior of the piece was concerned. The solution in this case was very simple, but the writer recalls other cases where a thorough knowledge of both the practical and theoretical side was required before the trouble could be located and remedied.

DETERMINING PROPER HEAT-TREATMENT TEMPERATURES.

We should keep in touch with the theory as much as possible, even while describing briefly some of the practical points to be observed in the heat-treating of steel. Let us suppose, for instance, that a new type of steel is to be hardened. If the analysis were known, it would help very much in estimating the correct hardening temperature, but it will be assumed that this is not known. If a critical-point outfit can be obtained, the critical points of the steel can be determined; then the critical temperature diagram, Fig. 1, will help us tell the correct temperature to use.

If an outfit of this nature is not at hand, samples of this steel, all cut to approximately the same size and shape, should be hardened in water at different temperatures, starting at 732 degrees C. (1,350 degrees F.) and increasing the temperature 15 degrees C. for each sample. The test pieces should then be broken by the same method so that the fractures will be comparative. All soft pieces may be disregarded and only the harder ones examined. Those with the finest grain may be separated from the others and should then be tested to ascertain which is the hardest. Nearly all shops possess either a scleroscope or Brinell hardness tester, but if these are lacking, a file may be used, although in this case it is a poor substitute.

It will probably happen that several pieces will look almost alike as far as grain size goes; in this case the hardest piece will represent the nearest approach to the correct hardening temperature. In other words, the piece that combines the finest grain with the greatest hardness is the one most perfectly hardened. If desired, when the temperature at which this piece was heat-treated has been noted, the test may be repeated by hardening, say, four more pieces at intervals of 5 degrees C. above and below the hardening temperature of the piece chosen by inspection. These pieces should be broken and tested as before, and the one showing the finest grain with the greatest hardness will represent the correct hardening treatment. The second experiment is performed only to locate the tem-

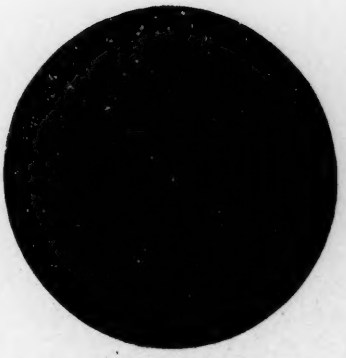


FIG. 2.—PHOTOMICROGRAPH OF STEEL CORRECTLY HARDENED.

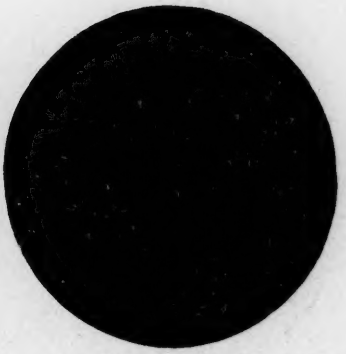


FIG. 3.—STEEL SHOWN IN FIG. 2 MAGNIFIED TO 400 DIAMETERS.

FIG. 4.—CAST STEEL BEFORE IT IS ROLLED OR HEAT-TREATED.

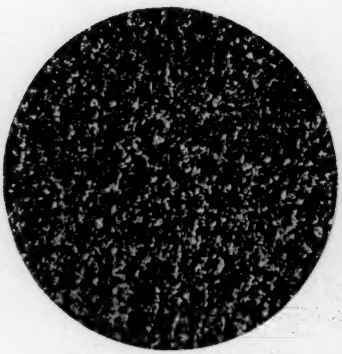


FIG. 5.—STEEL SHOWN IN FIG. 2. HARDENED AT TOO LOW A TEMPERATURE.

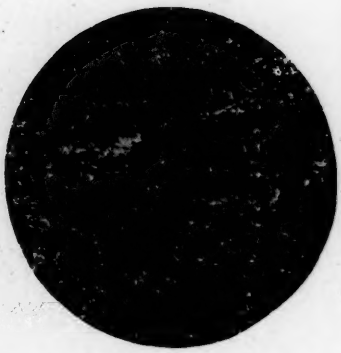


FIG. 6.—STEEL SHOWN IN FIG. 2, WHEN IT HAS BEEN OVERHEATED.

FIG. 7.—STEEL SHOWN IN FIG. 2, WHEN IT HAS BEEN BURNED.

FIG. 8.—HIGHER MAGNIFICATION OF STEEL SHOWN IN FIG. 7.

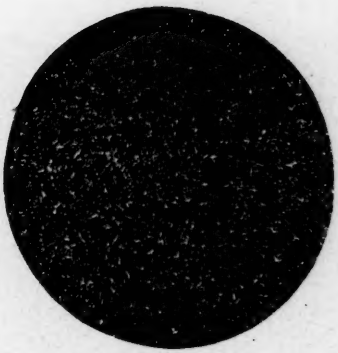


FIG. 9.—STEEL SHOWN IN FIG. 6 HEATED AT CORRECT TEMPERATURE.

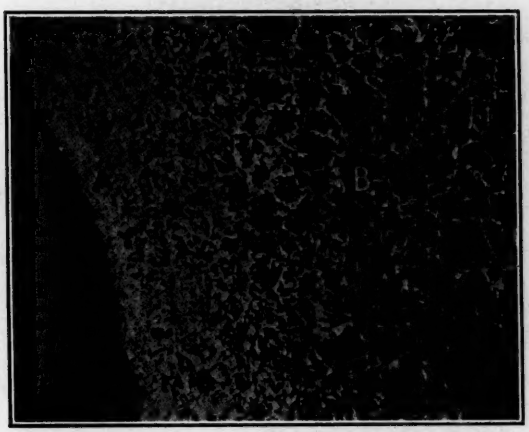


FIG. 10.—STEEL BADLY DECARBURIZED AT STEEL MILL.

perature a little more accurately than in the first experiment, where 15-degree intervals were used.

If the steel does not show any material increase in hardness at any of the temperatures tried, it is probably a low-carbon steel which, due to its carbon content, cannot be materially hardened. In this case, a file will not do for the hardness test, so a more delicate instrument must be used.

In many cases it will not be necessary to undertake the above experiment, as the manufacturer supplies the information regarding the hardening treatment. Where the chemical analysis is given, by referring to Fig. 1 the hardening temperature may be quite accurately estimated and a trial may be made to confirm it.

If the steel is an alloy steel, it must be borne in mind that in addition to the carbon content the special alloying element may influence the correct hardening temperature; for instance, nickel lowers the hardening point about 20 degrees C. for every per cent. of nickel in low-carbon steels containing less than 5 per cent. nickel; manganese also lowers it. Vanadium, chromium, and silicon have no marked effect.

Before proceeding further, we should understand what happened to the steel that was tested for its hardening temperature. Let it be assumed that it was about 0.50 per cent. carbon steel. All the test pieces that were heated up to a point below Ac_1, Fig. 1, remained soft on quenching. The first piece heated above Ac_1 and quenched showed an increase in hardness because the pearlite grains had been transformed into austenite and the quenching served to lock them, if not in this state, in an approximation to it which should be hard. As the different pieces were heated to successively higher temperatures through the range, more and more of the excess ferrite entered into the solid solution of austenite, so each of these successive pieces when quenched, therefore, was harder than the previous one. When those pieces that had been heated above the top of the range, where they are converted completely into a solid solution, were quenched, a maximum hardness was obtained. If these pieces were heated much farther above the top of the range and quenched, there would be no gain in hardness; rather the steel would be overheated, which results in a large grain size as already mentioned.

PHOTOMICROGRAPHS OF GOOD AND DEFECTIVE STEELS.

Fig. 2 is a photomicrograph showing the fine grain resulting from hardening a steel at the correct temperature. Fig. 3 shows the same steel magnified still more—to 400 diameters; yet it still appears very fine grained. The actual diameter of the spot of steel shown in Fig. 3 is only 0.008 inch; thus it will be seen that this photomicrograph shows a part of the steel about as big as a pin point. Fig. 4 shows a photomicrograph, of 100 diameters magnification, of cast steel before it was refined by the rolling process and by heat-treatment. The enormous difference in the grain size is apparent and serves to illustrate what changes can be wrought in steel by the correct treatment.

Fig. 5 shows a piece of the same steel as that shown in Fig. 2, which has been hardened at too low a temperature. The steel shown in Fig. 2 had a Brinell hardness of 578, while that shown in Fig. 5 had a Brinell hardness of only 248. To the eye the fracture of both might appear about the same, but the hardness test shows that the second one has not been heated to the top of the range, because it does not possess its maximum hardness. Fig. 6 is a photomicrograph of the same steel when it has been overheated; that is, heated above the top of the critical range farther than is necessary. It shows a coarser grain size, yet it is no harder than the steel shown in Fig. 2. Nothing has been gained, therefore, by overheating; in fact, the coarser grain size is a distinct disadvantage, because the metal is weaker and more brittle. Fig. 7 shows the same steel heated

to a temperature of 1,000 degrees C. (1,832 degrees F.), which is so much too high that it has resulted in burning the metal at some points. This is shown more clearly in the higher magnification, Fig. 8. These illustrations show the importance of finding the correct hardening temperature. If the steel is heated below this temperature to harden, the maximum hardness or strength is not obtained; if it is heated too far above this temperature, it is coarsened and its physical properties are impaired; and if the heating is carried still farther, the steel may be ruined by being burned.

If the steel with a very coarse grain, as shown in Fig. 6, is not heated to the top of its critical range, this coarse grain will not be entirely eliminated; but if it is heated to the top of the range and quenched, the coarse grain will be refined, unless it has been badly overheated and burned. Fig. 9 shows the same piece of steel as that in Fig. 6 after it has been refined by heating to the correct temperature.

However, if the steel has been previously burned or abused, it may be impossible to restore it to a fine-grained and strong state. For instance, unless the steel is remelted, no amount of heating can repair the cracks produced in the burned steel of Fig. 8. Fig. 10 also shows a case where no hardening temperature can bring the steel into a state of proper hardness, for the edges have been very badly decarburized by the steel mill. This is shown by the fact that the edge *A* appears lighter than the inner part *B*, because the former has less carbon bearing or pearlite grains.

APPROXIMATE TEMPERATURES FOR HARDENING CARBON STEELS.

Carbon, Per Cent	Hardening Temperature	
	Degrees C.	Degrees F.
0.10	885	1627
0.20	865	1589
0.30	845	1553
0.40	825	1517
0.50	805	1481
0.60	790	1454
0.70	780	1436
0.80	770	1418
0.90	760	1400
1.00 to 2.00	760	1400

It must be borne in mind that the temperatures given in the accompanying table are not to be arbitrarily followed, because other constituents in the steel, such as manganese, affect the correct hardening temperature. They are given to aid the beginner, so that he will know approximately where to start his test for the correct hardening temperature, if he knows the analysis of the steel. It is only possible to obtain the exact temperature by experience.

In practice, after the theoretically correct hardening temperature for a given steel has been obtained, it is advisable to heat slightly above this temperature to allow for any delay in getting the steel into the quenching bath or any other cause that might possibly allow the temperature of the steel to drop below the top of the critical range. If it drops one degree below this temperature, the piece will not possess the maximum hardness, while if it is heated ten degrees above the top of the range and the temperature should fall one or two degrees before the piece is in the quenching bath, the steel will still be above the top of its range and will not lose any of its hardness.

The alloy steels, especially those containing chromium, can be heated farther above the top of their critical range without impairing their physical properties by enlarging the grain than can the plain carbon steels. This fact should not be taken advantage of to "take chances" in the treatment, but should be considered as merely a point in their favor toward obtaining quality.

TIME NECESSARY FOR HEAT-TREATMENT.

The next point to be considered is the time required to heat the piece to the correct temperature and the length of time it should be held at this temperature. Generally, the rate of heating a piece to the hardening temperature is given very little attention. The excellent alloy steels of today can be heated quite rapidly without impairing their quality to a marked extent, and this fact is taken advantage of in commercial hardening on a large scale. It must always be borne in mind that when trouble does occur this point should be investigated, for too rapid heating may result in cracking the steel and impairing its qualities. This is especially true in the case of cheap grades of steel that have been hardened and are being rehardened without first being annealed or carefully preheated.

No fixed rule can be given for the rate of heating, as this point can be determined only by intelligent experience. When the piece has been heated to the proper temperature, it should be held at this temperature until sufficient time has been allowed for the complete grain transformation to take place throughout the whole piece. It was believed for some time that when the outside of the piece was of the correct temperature the interior portions were still much below this temperature, but recent experiments have shown that the interior temperature of the piece lags only a very little behind the exterior temperature. This must not be construed as a reason for not giving the piece time to "saturate," for this time must be allowed; its purpose is to give time for changes to be completed in the structure of the steel rather than to effect a complete temperature saturation of the whole piece.

A point of great practical importance and one that must not be overlooked in discussing the heating of the steel for hardening is the pyrometer and its location. A great many hardening furnaces have different temperatures at different parts of their heating ovens. The writer has observed cases where the pyrometer was in the rear of the furnace and read 1,420 degrees F. At the same time a piece located near the pyrometer was at a temperature of 1,400 degrees F., while a piece in the front of the furnace was at a temperature of only 1,380 degrees F. When quenched, a marked variation in the hardness of the two pieces was naturally noted. Wherever possible, the pyrometer should be as close to the piece being hardened as possible. If it is impossible to realize this condition, the hardener should make tests so as to know just what discrepancy exists between the temperature of the pyrometer and the piece; in other words, he should know the temperature conditions that exist in the furnace, so that even if they cannot be corrected they can be intelligently allowed for.

All pyrometer connections must be kept tight and clean and should be soldered wherever practicable. The pyrometer should be checked frequently with a master outfit kept solely for this purpose.

QUENCHING MEDIUMS.

Water quenching gives greater hardness than oil, and cold water and cold brine give still greater hardness. In choosing the quenching medium, we are therefore governed chiefly by the hardness desired and the amount of drastic quenching that the steel will withstand without cracking or severe distortion. A good many concerns use steels entirely unsuited for

the purpose intended, and get the necessary hardness by what might be termed "forcing"; that is, the steel is quenched in a very drastic medium, such as ice-cold water, which often results in cracking the steel. By using a better grade of steel or a steel of higher carbon content and quenching it less severely, as, for instance, in oil, the necessary hardness or strength may be obtained with a larger factor of safety against cracking and distortion of the steel. Some steels, however, are made to harden in water and others in oil, and the manufacturers of these steels give instructions to this effect. In the absence of definite information, the beginner can best determine this point for himself.

By removing the steel from the quenching bath before it has become entirely cold, the danger of cracking is greatly reduced, for this allows the steel to be slightly tempered. In some cases, excellent results may be obtained with steel that will crack if quenched entirely in water by placing it in water for a few moments and then removing it quickly and plunging it in oil, where the quenching may be completed. The length of time the work should be held in the water can only be determined by actual trial. Generally, it may be held there until the vibration felt along the handles of the "pick-up" tongs, due to the violent formation of steam around the piece, has almost or entirely ceased.

A very important point in quenching steel is to get the work into the bath as quickly as possible. Just as soon as the furnace door is opened, the inrushing air strikes the piece to be hardened and chills it. As the piece is removed from the furnace, this chilling effect becomes more pronounced. If the piece is one of complicated section, the smaller parts and edges may be materially affected. For this reason, the quenching bath should be located in the most advantageous spot for rapid work.

PACK-HARDENING AND TEMPERING.

Pack-hardening must not be confused with casehardening, where the work is also packed. In casehardening, the object of the packing is to give a pronounced case to certain portions of the steel; while in pack-hardening, the object is to heat the piece through thoroughly and to prevent the scaling or oxidation of the surface that will result, to a more or less extent, if the piece is heated in the open fire. The parts to be heated, therefore, are packed in metal pots or boxes with charcoal, or some other material that will not have a decarburizing effect upon the steel. The pots are then heated through at the proscribed temperature and held at this heat for the proper length of time, after which the pieces are removed and quenched.

It is very important to know just what temperature exists within the pots and how long they require to become heated through. Preliminary tests should be run with pyrometer ends inserted within the pots and enough data obtained so that an accurate idea exists as to how long each size pot should be heated with a given size load of steel within it. The old method of running wires, through holes drilled in the pot cover into the interior contents of the pot and withdrawing a wire at given intervals of time and noting its color as compared with that of the furnace walls may also be of value.

The proper tempering, or drawing temperature, as it is also called, depends on the use for which the steel is intended. If it must be intensely hard, it should be drawn, or tempered, at a very low temperature, or the operation may be omitted altogether. If a certain amount of toughness or ductility is desired, the steel must be tempered, unless it is a steel of such analysis, for instance a low-carbon steel, that it possesses enough ductility after being hardened only. Fig. 1 shows that, as we draw or temper a hardened steel, the solid solution (or its approximate state) is decomposed and constituents are formed that make the steel weaker and softer, but more ductile.

By tempering a straight carbon steel at about 400 degrees C. (752 degrees F.) it is changed into the troostitic state, unless it is a steel of such nature, or is hardened in such a way, that it contains some still lower constituent, even after hardening, such as sorbite. Troostite is much softer and less brittle than martensite, and is usually the most common constituent found in commercially hardened and tempered steels. By tempering at 600 degrees C. (1,112 degrees F.), which is generally considered annealing, a sorbitic structure that is still more soft and ductile than troostite is obtained.

After a steel has been correctly hardened, it should be drawn, experimentally, at different temperatures and tested to find out how it stands up for the particular application intended. Experience has shown the degree of heat required for tempering for quite a variety of purposes, and these temperatures may be obtained in numerous books on hardening. In practice, there are several methods of tempering hardened steel. The steel may be heated to the hardening temperature. The end desired hard is then plunged into the quenching bath until it is fairly well hardened and is then withdrawn, rubbed bright, and held until the heat of the unquenched part has crept down to the hardened end and heated it to the proper temper color, when the whole piece is plunged into the bath and quenched. In the hands of a skillful and experienced hardener, this method may be entirely satisfactory.

Another way of tempering is to heat the part in a bath of hot oil. Oil can be obtained that will stand a temperature of about 343 degrees C. (650 degrees F.). This is a very accurate way, as the exact temperature of the oil may be obtained by a thermometer and the piece is certain to be tempered uniformly all over. For temperatures between 260 to 583 degrees C. (500 to 1,100 degrees F.) a bath composed of equal parts of potassium nitrate and sodium nitrate may be used. Lead and sand baths are also employed for tempering. In every case, however, the same object is accomplished. The solid solution (austenite) or its approximation (martensite or even troostite) formed in hardening is decomposed into a lower constituent or lower combinations, such as martenso-troostite, troostite, troostito-sorbite, or sorbite itself. All of these have more ductility and less hardness than the constituents above them.

Some of the main factors in heat-treating steel and the underlying theory have now been considered. While it has not been possible to give exhaustive attention to every point, an effort has been made to show that there is a reason for each process and also what that reason is. Much progress remains to be made in the investigation of some of the causes of the changes observed in heat-treatment, but if each hardener will study the subject intelligently and forget the rule of thumb or the mysterious formula lent him by some kind acquaintance, he will soon acquire a knowledge that will help him to overcome obstacles and place his work upon a successful scientific basis.—"Machinery."

THE DESIGN OF AN OIL ENGINE.

By John F. Wentworth, Esq.,

Member, American Society of Marine Engineers and Naval Architects.

(*Abridged*).

The actual design presented here was made to fit the needs of the submarine—a light and reliable type of engine. If we were to consider a design suitable for a freighter, the matter of weight would not be so important, but the matter of reliability and low upkeep would be more important. The attempt has been made in presenting this design to give data and substantiating reasons for opinions where, perhaps, satisfactory

data cannot be presented, so that this paper may be taken as a basis and, if desired, other designs may be worked up in any well-organized drafting office.

In designing an engine there are several points to be settled upon as a matter of policy; from this point the design is not peculiar to the oil engine but rather is a straight matter of mechanical design. For instance, it has been suggested that the matter of bearing pressures is something to be settled by oil-engine experience. The opposite viewpoint is taken. It is felt that the bearing pressure in the oil engine should follow the same laws that are followed in other machines of somewhat similar action. For instance, we find the bearing pressure allowed in a locomotive is higher than that allowed in a triple-expansion engine. The reason is very easy to understand. In the locomotive the maximum pressure lasts for but a short part of the stroke, especially in single-expansion locomotives. On the other hand, in the compound marine engine the cut-offs are late and the maximum pressures extend over a much longer part of the stroke. Since the limiting point is the heating of the bearings, and since a machine which has an intermittent action has a greater time to dispose of the heat generated by the pressure, the maximum pressure per square inch may run higher. Engineering science has many examples of how this works out in practice, and it would seem that such question need not be of great moment to an organization skilled in machine design. This point is mentioned so that it may be understood that, in some, if not in many instances, data from work quite dissimilar to the design of the oil engine have been used. In such cases the application of the force is similar, even though the resulting design may differ very much from the completed oil engine.

The next point in a general consideration of the problem is the actual experimental data which have already been presented on this type of engine in the paper on "The Thermo-dynamics of the Marine Oil Engine" published in the "Transactions of 1914."

The third point consists in following certain deductions derived from experimental data and from a theoretical consideration of points in the design as affected by this experimental data.

The general features to be decided upon before the actual design is begun are as follows:—

1. The type, whether two-cycle or four-cycle.
2. The pressure of compression, and whether this pressure shall be constant or not.
3. The class of fuel to be used, and what steps shall be taken to insure its successful use.
4. The means for starting, and what, if any, steps are needed to make practical the particular type of starting settled upon.
5. The best arrangement of the air compressor.
6. The peculiar qualities necessary in fuel valve and fuel pump.

TYPE.

The type selected was four-cycle, and the engine designed for naval use. The Navy needs two speeds; one is a normal economical cruising speed, and the other a maximum possible speed, regardless of economy.

The four-cycle is unquestionably superior to the two-cycle, because it allows a greater variation. In a four-cycle engine by a simple matter of admission valve control any desired pressure of compression can be obtained. This is possible in the two-cycle only by making the normal pressure the low pressure. In other words, in the two-cycle the pressure of compression can be varied only by increasing the pressure of the scavenging air. Any regulation of the pressure by a reduction of the scavenging air below the normal would seem to be very difficult to obtain, that is, the normal compression should be that pressure which would result from hav-

ing a full cylinder of air at atmospheric pressure. It is quite difficult to see how any pressure lower than this is practical in the two-cycle engine. A higher pressure than this can be obtained by increasing the pressure of the scavenging air.

In the condition of the engine to-day there are many points which will remain a matter of opinion till settled by exhaustive tests. With a four-cycle engine these points in some instances can be settled by the use of such a type of valve action that the same engine can be run on different cycles or types of indicator cards. From such tests much valuable data could be obtained, and so, if for no other reason, it seems wise to adopt the four-cycle type.

Even the most ardent enthusiasts for the two-cycle type do not claim it comes nearer than nine-tenths of the efficiency of the four-cycle type.

Also this matter of simplicity must be considered. Is it simpler to have a given number of cylinders with a more elaborate system of valves in the head, or is it better to increase the number of cylinders by adding scavenging cylinders? The author holds that the four-cycle type is the more simple.

Notwithstanding this opinion, considerable attention has been given to the two-cycle type in order to make as many of the author's theories applicable to the two-cycle as is possible. It is hard to see how the weight of the four-cycle type can be reduced by making a two-cycle, and it seems that the simplicity of valve gear is more than offset by complications in the engine as a whole, and also that a great amount of flexibility is sacrificed by adopting the two-cycle type.

COMPRESSION.

Notwithstanding Diesel's theory, the compression pressure in the oil engine is primarily carried to the high point for the purpose of producing a temperature of ignition. The designer should consider this feature very carefully to see if a lower pressure cannot be used with better results.

We know that a pressure of compression of about 500 pounds will give a temperature sufficient to ignite the fuel spray. If this pressure can be reduced, the result will be lower-bearing pressures and reduced strains in the crank shafts. Since this means a lighter and more durable engine, it becomes of vital interest to know just how low a pressure of compression can be used. It has been fixed that from 450 to 500 pounds will be an average figure to use for the conventional type of engine. This pressure can be very greatly reduced by warming the cylinders, by steam for instance, before attempting to start the engine. The estimated pressure of compression will not vary from the actual pressure by any considerable amount. We know that a small four-cycle oil engine, when warm, will fire on a compression of 125 pounds. It is certain that under the same conditions—namely, after the engine has been warmed up by operation—the larger sizes will be able to run on pressures as low as 150 to 175 pounds.

This latitude may be objected to as being altogether too great to be tolerated in a design which is to be built. Just as the main bearings have to be scraped to fit the shaft, so it is possible to fit the cylinders with a proper clearance. This is not a very difficult matter; that is, in case of an error in estimate, no parts need be scrapped. There are several ways in which considerable adjustment can be made in the cylinder clearances. In the original engines put out by the American and British Manufacturing Company, the shaft was raised or lowered by means of wedges upon which the bearings rested. This seems to offer a little too much chance for meddling on the part of irresponsible parties, and it is suggested that the connecting rods may be lengthened or shortened to make the needed variation. In addition to this method, it is possible to get the same result by taking a chip from either the head or the cylinder, or a slightly

thicker gasket might be used in place of the original gasket. Even this change in the gasket might be objected to, while absolutely no objection can be found to taking a chip from the head of the cylinder in case the engine is found to have too great a clearance, and in case the clearance be not large enough to taking a chip from the connecting rod. In fact, with a marine type connecting rod, all the variation which is necessary can be made by either lengthening or shortening the rod. This matter of adjustment will not be a serious one, and when the designer understands that his estimated clearance can easily be altered, in case of need, without the waste of any machined parts, greater confidence should exist in his mind in regard to the final product. Having shown how the clearance can easily be altered to suit the needs of the engine, the necessity of wide experience is done away with.

The clearance will vary for the different types to be built. For instance, a small high-speed engine will run on a lower compression since the length of time is a minimum during which the compressed air at its maximum temperature is exposed to radiation losses under the worst condition. On the other hand, in a large and slow-speed engine, while the air at the end of the stroke has a longer time in which to lose sufficient heat to prevent ignition, the space in which the air is compressed is much greater, and hence the facilities for heat losses are less. It is very probable that there will not be a very great difference in the clearance which can be used. The time element in the high-speed engine is to a very great degree counteracted by the increased clearance of the larger engine.

The data presented in "The Thermodynamics of the Marine Oil Engine" prove conclusively that the full pressure of compression is not needed after the engine has been run long enough to get warmed up. Since the engine will warm up in a very short time under the fuel cycle, obviously it is important to decide how to bring the engine to a condition of design under which it will not only start but, after starting will be in the most ideal condition for operation under the fuel cycle. The engine built in large units must be jacketed when running on the fuel cycle. Vessels of any considerable displacement must have steam aboard for auxiliary purposes. Under these circumstances the most practical means for meeting this condition is to provide the water jackets with steam pipes, so that steam can be admitted to the cylinder jackets in order to warm the cylinders preparatory to the first explosion. The term "explosion" is used here with a purpose. Combustion and explosion differ chiefly in the time element. The explosion we usually associate with a rising pressure, and it is this rising pressure which is needed in the oil engine.

The ultimate efficiency of the engine is a combination of the thermal and mechanical efficiencies. The thermal efficiency depends upon the number of expansions. The mechanical efficiency depends upon the ratio of the mean effective pressure to the average pressure during the compression and expansion strokes.

The mechanical efficiency of the engine is the ratio of the brake horsepower over the indicated horse-power. The brake horse-power is the indicated horse-power minus the frictional loss. The frictional loss is a function of the pressure on the bearings and also certain losses in the auxiliaries. As far as the bearing losses are concerned or the main-engine friction, this is largely a function of the sum of the average pressure for the compression as well as for the expansion strokes. Without going into a detailed calculation it is evident that the sum of these two average pressures will be less in the low-pressure type than in the high.

For this reason it would appear that the low-pressure type has a greater power for the same piston displacement with an equal thermal efficiency and a greater mechanical efficiency. This seems to be the ideal condition sought for in the oil engine. It means a more powerful engine and at the same time reduced bearing pressures and stresses which will be reason for expecting a longer life and reduced upkeep.

Here is the proper place to take up the matter of the injection air. In the high-pressure type of engine about 10 per cent. of the net brake horse-power is consumed in the air compressor. Since the pressures are lower in the low-pressure type, the air for injection purposes can be reduced in pressure, which will reduce the loss in the air compressor, which in turn will give an additional gain in the efficiency of this type of engine.

For the marine engine, and especially for the naval engine, this is a most important feature. It is here that admission valve control comes in. In all present oil engines the speed is controlled by a variation of the fuel fed per stroke. This means that at reduced power the engine compresses an unnecessarily large amount of air per stroke. Unnecessary work means mechanical loss. For this reason it is proposed to reduce the amount of air compressed per stroke as we reduce the amount of fuel used per stroke. Under the Diesel school theories this is impossible. It has, however, been proven possible to a certain degree by the experiments of the author. The result is that no unnecessary frictional loss is occasioned by the compression of an unnecessarily large amount of air.

There is a theoretical gain in the low-power cycle which is independent of the mechanical gain. In the internal-combustion engine the pressure at the opening of the exhaust passages is still very high—from 20 to 30 pounds. Suppose we could collect this high-pressure working fluid and expand it in a secondary cylinder without loss from back pressure. This would result in an additional saving of a very considerable amount. On the proposed cycle this is accomplished. Let us assume that the admission valve is closed at two-thirds of the suction stroke. Then when two-thirds of the expansion stroke is completed a normal cycle is closed. On the low-power cycle the expansion is continued 50 per cent. further before the charge is exhausted. In this way the efficiency of the low-power cycle is greatly increased.

Besides this condition of economical low speed there comes the condition of extreme low speed which is only occasionally needed. Here economy is not a factor. The engine must run slow regardless of economy. The limit in slow speed comes from the loss of ignition temperature due to radiation. This can be overcome to a degree by controlling the circulation of the cooling water. As the engine slows down the work done per stroke is reduced, and consequently the heat which must be carried off is reduced. So far as is known the author is the first student of the internal combustion engine to suggest attaining a lower speed through a regulation of the temperature of the cooling water. This seems, however, to be important. There will never be a call for a prolonged operation at extreme low speed. In such cases as picking up a disabled ship at sea, or in similar emergencies, it is most important to be able to regulate the amount of cooling water fed to the jackets in order that the excessive radiation losses may be avoided. The means for accomplishing this result seem to be practical and to follow the commonly accepted laws of thermodvnamics, which apply to the oil engine as well as to the steam plant.

This is a subject of utmost importance. The oil engine has come to stay, and as the number of engines increases the inevitable result will be a rise in the price of the available fuel. The oil-producing interests have so far met the oil engine rather more than half way. There are fuels on the market known as "Diesel engine oils." These fuels are, if the author is not misinformed, distillates, that is, the refuse from the production of lighter oils is heated to a temperature above that at which the kerosene is evaporated and the resulting vapor is condensed. This gives a fuel which is fairly fluid, and which in a way is a by-product. Gasoline was originally a by-product and, like gasoline a few years ago, this fuel is at present very reasonable in price. It is not safe or wise to depend upon this condition continuing. The oil engine is doomed to be extremely limited in its application if this fuel is depended upon.

The fact that this fuel is a volatile product like gasoline, although vaporizing at a much higher temperature, is of no importance to the oil engine. All that is needed for the oil engine is a free flowing fuel. Even the presence of a considerable amount of non-volatile matter like the asphaltum residue seems to be of no disadvantage in this type of engine. The one essential is that the fuel must be free flowing. In this one great point Dr. Diesel stood out pre-eminently ahead of his time. He went so far as to propose the use of coal in a pulverized form in his engine. While the use of coal in the engine is of very doubtful value, the use of all grades of fuel oil is unquestionably practical. How to utilize these low grade fuels is very important.

We buy fuel by bulk, and it is practically true that all fuel oils have the same calorific value. The point to be considered is that when a heavy fuel is bought by bulk, although the calorific value per pound is somewhat less than for a high grade fuel, the calorific value per gallon is greater. Since we buy the fuel by the gallon this is what interests us.

Again, in order to protect the development of the engine against being checked by the rising price of the fuel oil, it is very essential to be able to use the same fuels as are used under the steam boilers. If this be done, since the oil engine has an efficiency of not far from three times that of the steam engine, the steam engine can be driven back to the use of coal before the oil engine can be seriously affected by the rise in the price of fuel.

Having shown the extreme desirability of the use of this low-grade fuel, next in order is the study of means for using it with good results. There seems to be one means which has already been proven in the steam plants, that is, to heat the slow-flowing fuels before feeding to the fuel pumps and fuel valves, and to do away with the jackets on the fuel valves while changing the design of the valves so that this can be done without serious trouble. As far as the operation of the engine is concerned, the heavy fuel, when thus treated, is in every way the equal of the gas oil. For this reason it becomes of the utmost importance that the treatment of the fuel, preparatory to spraying, be studied with the utmost care.

For all practical purposes it can be stated that the Baume of asphaltum base fuels will increase one degree for each increase of 15 degrees F. in temperature. With this change in Baume there is also a change in viscosity. This change is needed to make extreme low grades of fuel oil available. It will not be sufficient to simply heat the fuel oil; it must be kept hot till after the oil spray is formed. This means that the water jacket must be omitted from the design of the fuel valve, a step which has already been adopted by some engine builders. The only excuse for a water jacket on the fuel valve is to prevent combustion in the fuel valve. This can be prevented by a change in the design of the fuel valve, and better results will be had from this new type of valve.

In suggesting anything new in an engineering line the reason must be looked for to explain why this has not been done before. It is not safe to say that a reason does exist. Only too often the new idea has never been tried on account of a lack of inventive or imaginative ability rather than on account of too great an understanding of the obstacles confronting the designer. In this case there seems to be no obstacle in the way of building a universal oil engine. By this term is meant an engine which will use all grades of fuel without changing any of the parts of the engine. For instance, it might be possible to design an engine with several interchangeable fuel valves, so that as the type of fuel was changed a suitable type of valve would be available. This does not seem desirable or necessary. By controlling the application of heat to the fuel it seems very reasonable to believe that any grade of fuel will be satisfactory, and the only mechanical problem will be to design a fuel pump to handle the fuel and a fuel valve to produce the spray.

In his experimental engine the author had means for pumping up a

fuel pressure, by moving the block, which carried the lower end of the pump connecting rod, along a slide oscillated by an eccentric. When the engine was at rest, by moving this block from the position of no motion to the position of full stroke of the pump, a movement of the pump plunger was obtained. It was found that a slow motion of the plunger did not produce as good results as a very quick and sharp motion. This suggested that more ideal conditions could be obtained by designing a pump in which the fuel would be delivered by a blow on the plunger like that which would be given by the impact of a hammer. The only practical reason for this is that such a blow will start the check valves better. In a way the check valve is moved by the stream of fluid in the pipes, and the more rapid the motion of the fluid the more perfect is the action of the valves. No doubt the trouble which the author experienced was due to imperfect mechanical details, but if this modification will enable a more imperfect construction to produce the desired result, it cannot be out of place in a perfect mechanical device.

This is the first thing to be done with the engine, and has been the last to receive consideration. The reason is that the time under the starting cycle is very short, and the matter of efficiency is not worth considering. By efficiency is meant the thermal efficiency. It is a matter of efficiency whether the engine will start or not. It becomes a matter of efficiency if it be shown how the engine can be run for an appreciable time on the energy which can be stored in a compact form available for instantaneous use. This is a matter of practical utility rather than of thermal efficiency. That the matter of thermal efficiency does come into this consideration is only incidental.

The steam engine starts with an expansion stroke and has steam available for this stroke, so the starting feature is not a factor. In the internal-combustion engine the starting problem is a most difficult one. Not alone does the engine have to be accelerated but the acceleration has to impart sufficient energy to overcome inertia, do the work of the engine and at the same time produce the compression stroke.

Up to the present compressed air has been universally used for this purpose. The oil engine is still in its swaddling clothes, and many developments must be worked out before the engine can take its place in the world's work as a tried and reliable element. For the generation of heat on shipboard the use of steam is practically essential. Then it is recognized that air starting is not absolutely reliable, for air must be generated. This may be done by a small oil engine. The auxiliary plant in turn must be started by air. This condition has brought about the use of a small steam-driven air compressor, or an electric air compressor, supplied with current from a turbo-generator set driven by steam. This is mentioned to show that even now it is realized that steam, either directly or indirectly, is the only reliable means for starting the oil engine. This being the case, why not abandon the use of air altogether and start directly from steam? This feature was taken up in the previous paper on the oil engine, but not in detail. The data on Sheet C are presented to show the advantage in the use of steam. The effective energy in one cubic foot of water at a steam pressure of 200 pounds and of air at a pressure of 500 pounds are compared as well as the cost of producing this energy. This cost is a matter of small importance. However, bulk and first cost of apparatus are great factors. Energy for starting can be stored in the form of water in a tank or boiler having large volume. In storing high-pressure air it is not practical to store the energy in such bulk. It is rather necessary to store the air in individual bottles of relatively small capacity and great weight. For this reason it becomes very evident that more energy can be stored in the boiler than in the bottles for the same weight and space occupied.

As to cost, the air for starting is used for starting alone, and so there is a direct expense chargeable to starting alone. On the other hand, the water container can be used as an auxiliary boiler for steam-operated

auxiliaries. Hence this element cannot be properly charged to the cost of starting the engine.

As to reliability, there is no argument. Steam has proven its reliability, and the only advantage of the oil engine is its economy. The disadvantage of the oil engine is its lack of reliability and a certain feeling of doubt which exists with all new elements. The matter of unreliability can be done away with by the use of steam as outlined. There is a limit to the slowness with which the oil engine can run. Suppose it necessary to run below this limit, can air be depended upon? Apparently not. When the engine is running slow no great amount of power is generated, and this condition will not be needed for any length of time. The chief need of this extreme slow speed would seem to be in picking up a tow at sea or in making dock. In maneuvering to make dock there have been several instances where the air has given out and the navigator has had to stand by and watch his vessel strike in a way which must be particularly annoying to a good navigator. This feeling of uncertainty as to the condition of the air supply must do much to make the oil engine unpopular with the bridge force. On the other hand the use of steam drawn from the auxiliary boiler would do away with all this, and the navigator need not know but that his engine is a steam engine. He can come within maneuvering distance of the dock and then go on to the maneuvering cycle and bring his vessel up to the pier with absolute assurance. In order to make this possible in the oil engine it is necessary only to provide the engine with means for jacketing the cylinders with live steam. In fact, for the last case considered—that is, of maneuvering to come to a dock—it will be necessary only to cut out the cooling water and to depend upon the heat of the cylinder walls to prevent undue condensation. This is not a condition which need offer much difficulty, as the engine will simply follow the practice of the present steam engine.

The whole idea of this design is to reconcile the peculiarities of the oil engine to that thermodynamic practice which has been found to be the best in the use of the steam engine. The cylinder walls of the engine are jacketed, not to make them cold, but rather to prevent them from becoming too hot. It seems to be a necessary line of development to study this matter of temperature of the engine and to see that the engine be not reduced in temperature below that point which is desirable, just as in the steam engine the point of the best practical vacuum becomes a matter of judgment and not a case of obtaining the greatest possible vacuum regardless of cost.

The first question to be settled in the design of an oil engine should be what pressure of compression shall be selected for the full-compression pressure. The writer is intolerant of restraint. The oil engines in use to-day have jacketed heads and use a pressure of compression of 500 pounds. This is used because Dr. Diesel used it. Dr. Diesel took the suggestion of Du Rochas and made it practical by fixing the pressure of compression. How he arrived at this is unknown. Too great credit cannot be given to Dr. Diesel for his progressive spirit. Too great condemnation cannot be given to his imitators who, with additional data on hand, still copy the cycle which Diesel hit upon as a result of his groping around in the dark. In this submarine engine the compression at full load was fixed at 200 pounds. The clearance was found to be 15.6 per cent., and the height of the clearance to be 2¼ inches. This design was worked out, not alone to illustrate to the Navy Department certain ideas in oil engine construction, but also along lines upon which the author was willing to stake his reputation. In order to start this engine steam would have to be supplied to the water jackets. This is undesirable for such a small engine, but the submarine of to-day is not the submarine of to-morrow. When the offer was made to design a submarine engine reference was made to such an engine as would be needed in the large ocean-going submarines in which a speed of 25 knots would be obtained. These submarines have been de-

signed and the speed reduced to 19 knots because of the inability to obtain the above speed. To produce steam needed to warm this engine will not present any difficulty, and the engine will be greatly improved by an increase in power and an increase in mechanical efficiency. As soon as the engine is firing, water will be needed in the jackets and the engine will then follow the conventional Diesel in its cycle.

Extreme emphasis has been laid on the fact that this design is not claimed to be mechanically perfect. If put into the hands of a first-class drafting force and the several details gone over with great care, the result would be a great improvement. But the contention is made that the designer should keep his hands off the general arrangement.

The motor ships run by four-cycle engines are for the most part reversed by swinging the cams clear of the cam shafts while the cams are being moved along the cam shafts. This is needed for a multi-cylinder engine, because otherwise one or another set of cams would always be prevented from moving by coming in contact with its roller. This difficulty is remedied in the motor-ship, as has been stated, by swinging all rollers clear of the cams during a change of cycle. For a compact engine this arrangement is rather clumsy. The same result obtained by this swinging movement must be obtained. In this engine it is obtained by a modification of the roller and roller fork. The rollers are carried on a long pin with considerable play on each side of the rollers. The rollers are held in a central position by means of balance springs. With this design, in changing from one cycle to another, those cams which interfere with a roller will simply push the interfering roller to one side and the other cams are not prevented from attaining their new position. The displaced rollers take their normal position as soon as the cams have rotated enough to allow this action. In making the change from one cycle to another in a 6-cylinder engine there will be normally 24 cams bearing on the cam surfaces. These cam rollers must be held down by strong springs in order to assure prompt action of the valves at high speeds. To move the cams along the cam shaft would naturally call for the expenditure of considerable effort. In order to make this action easier and the maneuvering of the engine more prompt a provision is made on the cam boxes whereby all cam rollers may be simultaneously lifted about 1/32 inch. This will be enough to take the pressure off the cams and yet not enough to interfere with the valve action.

The start of this engine is obtained by this same lever. The tappet rod which operates the starting valve is lifted by a further motion of this same lever which will open all the starting valves. This sets the engine in operation, and the valve timing determines the proper direction of rotation. This is done by having a longer and different shaped cam acting on the starting valve tappet. This will be wasteful of starting energy, but the waste will not be excessive and the desired effect will be an instantaneous response to the reverse lever.

The desired operation of the admission valve is obtained by means of varying the timing of the admission valve cam and by making the valve automatic as far as opening is concerned, and mechanically operated as far as closing is concerned. This is done by having the valve cam simply release the admission valve. When this is done the valve stays open as long as the suction stroke is in progress. A partial suction stroke may be obtained by advancing the admission valve cam. In this way the valve opens but not until the suction stroke of the piston begins. On the other hand, when the valve cam completes its operation the tappet rod falls and closes the admission valve. The author is of the opinion that no novelty exists in this method of operation.

This is a most important matter and this design is unquestionably superior to the submarine engines at present installed in the United States submarines. If we divide the method of fuel feeding into two classes we shall have the present class and the proposed class. We may call these

two classes the Diesel type and the Wentworth type. As far as the author knows he is the first to propose this modification. Sheet D shows this difference. In the Diesel engine the operation of the fuel pump is slow. The plunger is practically moving all the time, but the stroke is so short that the speed of the plunger is very slow. The author has found by actual experience that a fast-moving plunger will give better results than this slow-moving plunger. The result is the fuel pump, as shown in detail. The feature of this pump, is a floating piston. This piston or plunger is lifted in the delivery stroke by a lever operated by a cam. The plunger is returned by the springs at the bottom of the plunger. The plunger is at all times kept against the lever by means of the spring. The motion of the lever determines the stroke of the plunger, and the movement of the lever is controlled by an eccentric shaft. With full eccentricity of the shaft up the lever is held so that the cams clear and no fuel is delivered. With the shaft turned an angle of 180 degrees the maximum stroke of the plunger is obtained. Any desired position may be obtained for the shaft between these two extreme positions, and so the amount of fuel delivered may thus be governed. The feature of this pump is not only to deliver exactly the fuel required, but to deliver this particular amount and at just the moment it is needed. By varying the position of this set of cams the action of the fuel pump may have a different timing. In place of a long suction and delivery stroke the two strokes are very sharp and quick, the plungers being idle the rest of the time.

Now referring to the plan of the fuel valve it will be seen that this valve really is a double valve. The air for injection has one seat and the fuel comes through another seat. The lift of the fuel plunger should be timed to act with the opening of these two valves so that the fuel will be delivered by the pump directly into a stream of moving injection air. In this way the system can take care of any grade whatever of fuel. It has been pointed out that the trouble with most fuel valves is that the fuel is delivered to the fuel valve and allowed to come to rest, and then the air has to get in and around the fuel so as to deliver it in small portions and not as a whole. This design of fuel valve absolutely does away with any difficulty along this line.

Objection may be made to the care needed in timing the two elements—the fuel pump and the fuel valve. No reply is needed to such an argument. It is no more difficult than to time the spark on a gas engine. Another feature of the fuel valve is that the fuel starts after the air has started, and stops while the air valve is still open. The result is that the fuel valve should give no trouble with clogging—it is cleared by air after each delivery stroke.

Attention is called to the fuel heating tank. Here the fuel is heated by means of a coil of steam pipes within a tank. By this means a heavy, sluggishly moving fuel may be made to flow as freely as the present gas oil. The result is a saving of 50 per cent. in cost. With the particular type of fuel valve shown, no trouble should be experienced in getting this fuel too hot, beyond gasifying it, which would give an unequal action of the engine if carried to excess.

All engines at present used in the United States submarines are said to use this comparatively high-priced gas oil. The engines shown in this design will run from the start in any grade of fuel, and it is thought that no other design of marine oil engine will do this. The author is pleased to state that an engine has actually been operated in this manner on low-grade California oil.

In the paper on "The Thermodynamics of the Marine Oil Engine" a great deal was said about the use of steam for starting the oil engine. There is no question about the advantage of steam in connection with the oil engine. At least the author has studied the matter from all viewpoints and can find no fault with the use of steam. Under these circumstances

it is no more than right that some consideration be given to the type of steam generator which would give satisfaction in this service.

The steam generator must have a large water body. There is one disadvantage to this—that it will take a considerable time to get this water body up to the desired temperature. This is true, but we are not handling a toy or quick-acting device. This type of engine is designed for actual service conditions seven days in the week. By this is meant that the engine is designed to be kept ready for use seven days in the week, even if used but once a month. While it would take a long time to get this water body up to temperature it would take but little energy to keep it there. For the navy this means that the watch on duty should see that this steam pressure be kept up. The fuel expenditure will be small.

The probable change in the submarine will be to enlarge it to fit it for duty as a commerce destroyer on the high seas. For men to work at their best they must be kept reasonably comfortable. This does not mean to live in cold damp quarters. The vessel must be kept heated and the auxiliaries run from the waste heat energy of the present-day engine. It is strange that this had not been taken up before, but because it has not been is no reason for further delay.—"The Steamship."

S. S. *EDWARD LUCKENBACH.*

A notable addition to the fast-growing mercantile fleet of the United States has recently been made in the delivery of the freight steamer *Edward Luckenbach.* This vessel is the first of five now under construction for the Luckenbach Steamship Company of New York and San Francisco, by the Fore River Shipbuilding Corporation of Quincy, Mass.

The *Edward Luckenbach* was designed to the owners' special requirements for coast-to-coast and overseas trade, and has an adaptation of the cruiser stern.

This vessel is of the shelter-deck type, with large cargo hatches, wide spaced hold stanchions, wide-spaced frames, and no side stringers, and has been constructed under the special survey and to meet the full requirements for hull, equipment and machinery of both the American Bureau of Shipping and Lloyd's Register of Shipping, to obtain the highest classification ratings, with freeboard assignment, in the records of both societies.

The *Edward Luckenbach* is of the following dimensions:

Length over all, feet and inches	456–5
Length, b. p., feet and inches	437–6
Breadth, moulded, feet	57
Depth, moulded to shelter deck at side, feet and inches	41–6
Gross tonnage	8,151
Net tonnage	6,141
Panama Canal, gross tonnage	8,543
Panama Canal, net tonnage	6,426
Loaded, draught, feet and inches	31–8
Deadweight, tons	12,250
Height, between shelter and upper decks, feet and inches	9–6
Height, between upper and second decks, feet	9
Fuel oil capacity, tons	2,131
Capacity of cargo holds, cubic feet	584,270

There are three continuous steel decks running the full length of vessel, named shelter, upper and second, respectively. The second deck beams are in excess of classification societies, requirements, and with a 'tween deck height of 9 feet 6 inches, this vessel readily adapts itself for use as a transport or for carrying cattle.

The double bottoms are constructed on the alternating solid and bracketed-floor principle, and, with the forward and after peak tanks, are arranged for the stowage of fuel oil. The fuel oil settling tanks and the fresh and boiler-feed water tanks are constructed in the 'tween deck spaces on the sides of the machinery casings.

The arrangement of masts, king-posts and deck winches has been made with a view to the rapid handling of general freight. The foremast has one 30-ton and four 5-ton booms, the mainmast one 20-ton and four 5-ton booms, and each of the eight king-posts has two 5-ton booms. The deck winches, constructed by the Hyde Windlass Company to the special requirements of the owners, are 24 in number, 20 being single-geared and 4 compound-geared, all with cylinders 8¼ by 10 inches.

The steering machinery, also furnished by the Hyde Windlass Company, is of the Brown steam tiller type, with cylinders 9 by 12 inches, and operated by telemotor from the pilot-house and flying bridge and by shafting from double-wheel hand gear stand on top of after deck-house.

PROPELLING MACHINERY.

The propelling machinery is amidships, and consists of a Curtis turbine driving the propeller shaft through a double reduction gear at a designed normal speed to deliver 4,000 shaft horse-power at 90 revolutions of the propeller shaft. The turbine is fitted with overload nozzles so that the machinery will develop 4,500 shaft horse-power, with a propeller speed of 93 revolutions per minute. The turbine is designed to operate with a steam pressure of 190 pounds in the steam chest and against a vacuum of 28½ inches, with the astern turbine capable of delivering to the shaft about 50 per cent. of the power of the ahead turbine. The rotor is so arranged that the ahead and reverse wheels are carried on one shaft and operate in the same casing, the main exhaust being common to the ahead and reverse. The turbine is very substantially built of high-grade steel, the blading being of monel metal, of heavy proportions, and the revolving element is designed so that it can easily stand an overspeed of 20 per cent. above the full speed revolutions.

The total reduction of speed between the turbine and propeller shaft is 37 to 1. The reduction is taken in two steps, the system employed being what is known as the double-drive type, that is, the pinion connected with the turbine engages with two gears, one diametrically opposite the other, thereby eliminating pressure on the pinion bearings and insuring perfect tooth contact between the pinion and the wheels. Each of the two wheels is coupled to a pinion, which in turn drive the main gear coupled to the propeller shaft. The high-speed pinion is directly connected to the turbine shaft by means of a flexible coupling so arranged that any motion in the gears or pinions will not be transmitted to the turbine shaft. The gears are double helical, and are arranged in one plane so that bearings and wheels can be easily examined and all parts readily kept in alignment. The turbines and gears were built by the General Electric Company, the gears being of the laminated type manufactured under the Alquist patents.

The propeller is 19 feet diameter, with the blades set at a pitch of 16 feet. The pitch, however, is adjustable from 15 to 17 feet. The propeller blades are of manganese bronze and the hub of cast-iron fitted to the propeller shaft with a taper and key and held by a forged steel nut.

Steam is supplied by three Scotch boilers, each 16 feet 6 inches inside diameter, and 12 feet 3 inches long over the heads, with a heating surface of 3,546 square feet, each boiler having four Morrison corrugated furnaces, 42 inches inside diameter. The boilers are designed for a pressure of 195 pounds per square inch, and work under the heated system of forced draft, for which one Sturtevant multivane blower fan with two engines is located in the engine room. The boilers are arranged to burn fuel oil on the Dahl system, the necessary pumps, filters, heaters, &c., being arranged in the fireroom.

The main condenser of special construction for turbine work is placed on the port side of the engine, having a cooling surface of 5,850 square feet. The air pump, furnished by Blake & Knowles, is of the independent wet and dry duplex beam type, the necessary cooling water being supplied by the main centrifugal pump, having an 18-inch suction and discharge and driven by a 10 by 9 inch single-cylinder vertical engine of the enclosed type and supplied with forced lubrication.

The electric plant consists of two 20-kw. General Electric generator sets, driven by reciprocating engines. There is a 2-ton Brunswick refrigerating machine and a Griscom Spencer evaporating machine of 35 tons capacity.

TRIAL TRIP DATA.

Time.	Revolutions.			S.H.P.		Fuel Oil.			Water Turb. only.	
	Counter Reading.	Period.	R.P.M. Period.	T. Meter Read. 4,834.	Period.	Meter Read.	Lb. per hour.	Lb. per S.H.P.	Lb. per hour period.	Water rate period.
12.00	125,436	...	...	70	...	900	...	...	...	...
12.10	126,282	846	84.6	78	3,190	967	...	...	...	...
12.20	127,135	853	85.3	76	3,140	1,075	...	...	...	...
12.30	127,999	864	86.4	81.5	3,400	1,114	...	...	...	...
Ave.	...	...	85.4	...	3,265	...	3,200	.980	39,700	12.17
12.40	128,876	877	87.7	85.0	3,600	1,178	...	...	...	...
12.50	129,764	888	88.8	85.5	3,670	1,268	...	...	...	...
1.00	130,658	894	89.4	89.5	3,865	1,362	...	...	...	...
1.10	131,547	889	88.9	86.5	3,715	1,422	...	...	...	...
1.20	132,440	893	89.3	88.5	3,820	1,522	...	...	...	...
1.30	133,346	906	90.6	89.5	3,920	1,610	...	...	...	...
Ave.	...	...	89.1	...	3,787	...	3,710	.980	42,600	11.24
1.40	134,261	915	91.5	90.0	3,980	1,715	...	...	...	...
1.50	135,189	928	92.8	94.0	4,210	1,812	...	...	...	...
2.00	136,122	933	93.3	94.0	4,235	1,902	...	...	...	...
2.10	137,055	933	93.3	95.0	4,280	1,986	...	...	...	...
2.20	137,986	931	93.1	94.0	4,230	2,065	...	...	...	...
2.30	138,916	930	93.0	92.5	4,150	2,173	...	...	...	...
Ave.	...	...	92.8	...	4,181	...	4,210	1.007	46,650	11.17
MEASURING WATER FOR ALL PURPOSES.										
2.30	138,916	...	...	...	...	2,173	...	...	...	...
2.40	139,844	928	92.8	96	4,300	2,254	...	...	...	...
2.50	140,773	929	92.9	96	4,310	2,352	...	...	...	...
3 00	141,702	929	92.9	97	4,350	2,450	...	...	...	...
Ave.	...	...	92.9	...	4,320	...	4,140	.960	54,800	12.67
Overload			98	110.5	5,230	With 2 overload nozzle valves open.				

Draft = 10 ft. 4½ in. forward, 18 ft. 9 in. aft. Displacement = 7,300 tons.
Speed by log, 14 knots.
Fuel oil, 18,170 B.T.U. per lb.
0·894 Sp. Gr. at 60 F.

ENGINE ROOM.

Period.	Pressures lb. gauge.			Vacuum.	Bearings.		Temperature of		
	Main steam.	Steam chest.	Oil.		For'd.	Aft.	Injection.	Discharge.	Room.
12.00 to 12.30...	176.8	137.3	8.8	28.75	103.5	85.5	44	69	74
12.30 to 1.30...	179.4	159.3	12.0	28.75	103.3	98	45	64.7	75
1.30 to 2.30...	191.0	176.1	17.0	28.75	103.0	117.7	45	66.3	79
2.30 to 3.00...	194.0	179.0	17.0	28.90	102	116.5	43.5	65.0	78.5
Overload	195	170	28.75						

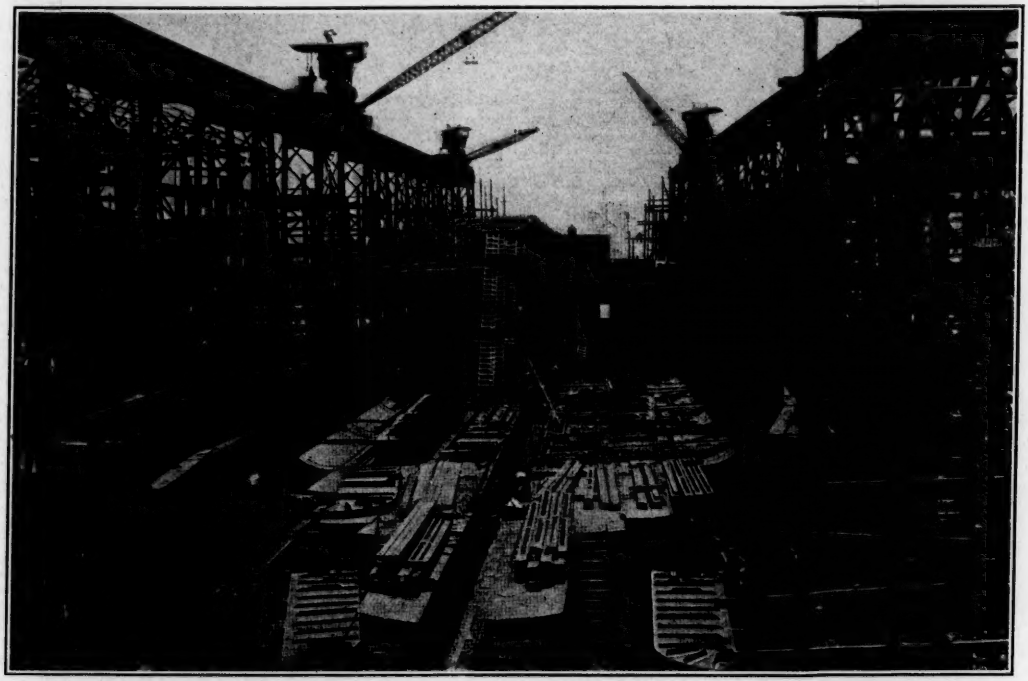
FIRE ROOM.

Period.	Pressures lb. gauge.			Fuel oil at burners.	Temperature of			Air press. in.
	Main steam at boilers.				Fuel oil heater.			
	Stbd.	Center.	Port.		Inlet.	Outlet.	Room.	
12.00 to 12.30...	185.5	182.8	183.5	121.7	103.7	247.8	96.7	.9
12.30 to 1.30...	182.1	179.3	179.9	146.4	104.0	248.2	92.9	.87
1.30 to 2.30...	192.4	189.2	189.4	165 7	99.3	253.6	88.9	.93
2.30 to 3 00...	196	192.5	192.5	158.8	97.5	254.0	86.0	.98

The *Edward Luckenbach* was taken out on trial on November 28, the ship being in light condition, with measuring tanks and pumps installed so that the water measured by the main turbines alone, and the turbines and auxiliaries combined, could be carefully ascertained.

A torsion meter was fitted on the propeller shaft, the shaft having been calibrated together with the torsion meter before being installed on board. The oil used on the trial was heavy Mexican, which on test showed only 18,170 British thermal units. The whole of the machinery operated very satisfactorily, and after anchoring at Boston Harbor for a couple of hours the vessel proceeded to New York, the trip from Boston to New York occupying about 22 hours.

The preceding table gives data taken on the trial trip. The results are considered very satisfactory as, at normal designed rate of operating, the water rate of the turbines was only 11.2 pounds per shaft horsepower and the water rate of the combined turbine and auxiliary machinery was but 12.67 pounds per shaft horse-power of the main engines.—"Shipbuilding and Shipping Record."



THE CONSTRUCTION OF OIL TANK STEAMSHIPS HAS BEEN A PROMINENT FEATURE OF THE YEAR'S WORK IN AMERICAN SHIPYARDS. OIL TANKER UNDER CONSTRUCTION AT THE UNION IRON WORKS, SAN FRANCISCO.

SHIPBUILDING IN THE UNITED STATES IN 1916.

OVER 500 MERCHANT VESSELS OF 1,718,338 GROSS TONS NOW UNDER CONSTRUCTION—263 MERCHANT VESSELS OF 491,161 GROSS TONS BUILT IN 1916.

According to reports sent to "Marine Engineering" from practically every shipyard in the United States, there are at the present time under construction in the private shipyards of the country 506 merchant vessels of 1,718,338 gross tons. In addition to the merchant work the private shipyards are building 53 naval vessels aggregating 174,021 tons.

The output of the private yards during 1916 amounted to 263 merchant vessels of 491,161 gross tons and 15 naval vessels of 45,925 tons.

The character of the output is significant, as it consists principally of sea-going cargo boats and oil tankers. In 1916, 47 sea-going freight steamers of 186,493 gross tons were built and 27 oil tank steamships of 180,156 gross tons. In the work now on hand there are 158 freighters of 794,963 gross tons and 64 oil tankers of 450,263 gross tons. Only a small portion of the output consists of passenger ships, and these are for coastwise service or for use on inland waterways.

As shown in Tables I and II, which give the number tonnage and horsepower of both merchant and naval vessels built by American shipbuilders in 1916, and corresponding figures for the work now under construction or contracted for, it will be seen that the largest output of merchant tonnage in 1916 was from the Union Iron Works, San Francisco, Cal., while nearly an equal amount was produced on the Great Lakes by the American Shipbuilding Company. Of the shipbuilders on the Atlantic coast the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., produced the largest amount of merchant tonnage.

TABLE I.—MERCHANT CONSTRUCTION.

SHIPS BUILT IN 1916.

	No.	Gross Tons	H. P.
Union Iron Works	9	65,293	24,400
American Shipbuilding Company	19	64,000	25,200
Newport News Shipbuilding & Dry Dock Co.	7	51,737	20,300
Great Lakes Engineering Works	14	35,214	19,175
Wm. Cramp & Sons' Ship and Engine Building Company	7	34,996	16,800
Fore River Shipbuilding Corporation	5	33,082	15,300
Bethlehem Steel Company (Maryland)	8	31,952	18,400
New York Shipbuilding Company	9	27,474	10,150
Harlan & Hollingsworth Corporation	5	18,835	20,434
Moore & Scott Iron Works	2	14,200	4,800
American Bridge Company	31	12,129	.....
Toledo Shipbuilding Company	6	11,020	1,920
Skinner & Eddy Shipbuilding Corporation	2	11,460	5,000
Chester Shipbuilding Company	2	11,000	.....
Baltimore Dry Dock & Shipbuilding Co.	4	10,603	4,500
American Car & Foundry Company	25	10,567	.....
Kelly-Spear Company	6	4,035	.....
Seattle Construction & Dry Dock Co.	1	4,000	2,500
Wilmington Iron Works	2	4,000	1,400
Mathews Shipbuilding Company	3	2,920	1,940
Peterson Shipbuilding Company	3	2,800	.....
St. Helens Shipbuilding Company	3	2,783	1,040
Kruse & Banks	2	2,450	2,200
Manitowoc Shipbuilding Company	6	2,324	3,400
M. M. Davis & Son	5	2,250	1,025
E. J. Tull	4	1,940	45
Wilson Bros. Shipyard	2	1,920	1,700
G. G. Deering Co.	1	1,891	.....

	No.	Gross Tons	H. P.
Percy & Small	2	1,620	.....
Staten Island Shipbuilding Company	5	1,511	2,350
Tank Shipbuilding Company	3	1,257	.....
Howard Shipyards Company	3	1,225	1,075
John Wilson Shipbuilding Company	5	1,110	210
Bean Shipbuilding Company	1	1,100	360
McKie Shipbuilding Company	1	922	1,600
Barbare Bros.	14	814	1,400
Arthur D. Story	4	738	400
Francis Cobb Company	1	578	800
F. S. Bowker & Sons	1	565	.....
Sawyer Bros.	1	435	.....
Bath Iron Works	3	399	8,500
Townsend Marine Railway & Construction-Company	2	280	420
Dubuque Boat & Boiler Works	3	234	105
W. F. Stone	4	220	282
W. & A. Fletcher Company	1	210	600
William H. Baldwin	2	197	40
Cowles Shipyard Company	2	70	125
Mathis Yacht Building Company	8	....	.....
John H. Mathis Company	4	....	.....

MERCHANT VESSELS NOW UNDER CONSTRUCTION.

	No.	Tons Displ't	H. P.
Union Iron Works	26	177,128	73,600
New York Shipbuilding Company	17	104,501	46,650
Newport News Shipbuilding & Dry Dock Co.	13	94,529	33,100
Bethlehem Steel Company (Maryland)	15	93,007	35,950
American Shipbuilding Company	28	92,900	36,900
Pennsylvania Shipbuilding Company	10	92,000	.....
Chester Shipbuilding Company	14	71,500	.....
Fore River Shipbuilding Company	10	66,544	30,400
J. F. Duthie & Co.	7	61,600	17,500
Williamette Iron Works and Northwest Steel Company	7	61,600	.....
Harlan & Hollingsworth Corporation	13	60,744	33,830
Standard Shipbuilding Company	8	58,400	20,000
Great Lakes Engineering Works	12	54,150	19,900
Wm. Cramp & Sons' Ship and Engine Building Company	8	53,810	36,000
Skinner & Eddy Shipbuilding Corporation	9	53,580	22,500
Columbia River Shipbuilding Corporation	6	52,800	15,000
Seattle Construction & Dry Dock Co.	10	49,850	27,500
American Bridge Company	92	47,616	.....
Tank Shipbuilding Company	27	47,500	.....
Moore & Scott Iron Works	5	45,300	12,200
Texas Company	4	40,000	.....
Baltimore Dry Dock & Shipbuilding Co.	9	29,745	12,700
Toledo Shipbuilding Company	7	19,250	9,500
Manitowoc Shipbuilding Company	9	16,856	8,980
Pusey & Jones Company	13	13,300	.....
Staten Island Shipbuilding Company	6	12,600	.....
Gray's Harbor Shipbuilding Company	5	12,500	3,500
McEachern Ship Company	6	10,200	3,360
Washington Shipping Corporation	4	10,000	1,920
Palmer Shipbuilding Company	4	10,000	.....
M. M. Davis & Son	8	9,900	.....
Brunswick Marine Construction Corp.	5	9,550	.....
American Car & Foundry Company	10	8,892	.....

	No.	Tons Displ't	H. P.
St. Helens Shipbuilding Company	4	6,200	2,240
Coastwise Shipbuilding Company	6	5,820	.....
Columbia Engineering Works	5	4,950	2,400
Peterson Shipbuilding Company	4	4,700	1,250
Bean Shipbuilding Company	3	4,000	.....
W. F. Stone	2	3,600	2,000
Mathews Shipbuilding Company	2	3,600	1,280
Hanlon Dry Dock & Shipbuilding Co.	1	3.500	.....
Kelly-Spear Company	2	3,204	.....
Atlantic Coast Company	2	3,200	.....
Merrill-Stevens Company	6	3,175	1,800
Percy & Small	2	3,000	300
Kruse & Banks	3	2,400	2,000
Francis Cobb Company	3	2,340	.....
Tampa Foundry & Machine Company	1	2,000	1,000
Spedden Shipbuilding Company	5	1,645	3,650
W. & A. Fletcher Company	2	1,612	6,400
G. G. Dearing Company	1	1,600	.....
Peninsula Shipbuilding Company	2	1,600	.....
Sandifer-Clarkson Shipbuilding Company	1	1,600	480
Georgia Shipbuilding Company	1	1,350	.....
G. A. Gilchrist	1	1,200	.....
Sharpstown Marine Railways Company	1	1,150	.....
Hillyer, Spearing & Dunn	1	1,100	.....
Dubuque Boat & Boiler Works	2	1,0 0	200
Townsend Marine Railway & Construction Company	1	1,000	.....
Bath Iron Works	1	910	8,400
E. J. Tull	1	900	.....
Howard Shipyards Company	3	800	925
Arthur D. Story	3	765	1,150
Brunswick Shipbuilding Company	1	750	.....
B. P. Lanteri	1	739	300
F. S. Bowker & Sons	1	550	.....
Sawyer Bros.	1	500	.....
Jas. Rees & Sons Company	1	460	.....
William H. Baldwin	1	101	.....
Cowles Shipyard Company	1	95	350
Barbare Bros.	30	....	1,500
John Wilson Shipbuilding Company	32	....	.....

TABLE II.—NAVAL CONSTRUCTION.

VESSELS BUILT IN 1916.

	No.	Gross Tons	H. P.
Newport News Shipbuilding & Dry Dock Co.	1	31,400	32,400
Fore River Shipbuilding Corporation	10	7,160	51,000
Wm. Cramp & Sons' Ship and Engine Building Company	3	3,290	54,000
Bath Iron Works	1	1,075	16,000

NAVAL VESSELS NOW UNDER CONSTRUCTION.

Newport News Shipbuilding & Dry Dock Co.	3	97,200	102,000
New York Shipbuilding Company	2	32,640	35,000
Fore River Shipbuilding Corporation	21	16,630	224,000
Seattle Construction & Dry Dock Co.	5	9,935	20,000
Bath Iron Works	6	6,776	132,800
Lake Torpedo Boat Company	8	4,400	.....
California Shipbuilding Company	6	4,190	1,300
Wm. Cramp & Sons' Ship and Engine Building Company	2	2,250	37,000

Considering the work now on hand, it is seen that again the Union Iron Works leads with the greatest volume of merchant tonnage under construction, with the New York Shipbuilding Company, Camden, N. J., second. A noteworthy feature of these returns is the fact that not only are the old established yards filled practically to capacity, but also many of the newly established yards have a volume of tonnage booked equal to, if not greater than, the normal output of the older yards in previous years.

Concurrent with this report there has been published in the annual report of the Commissioner of Navigation, Department of Commerce, figures covering the output of American shipyards for the fiscal year ending June 30, 1916. According to this report 937 vessels of 325,413 gross tons were built and officially documented in the United States, compared with 1,157 vessels of 225,122 gross tons for the previous year. These figures cover only the vessels built and put into commission during the year, and by no means indicate the remarkable activity in all American shipyards, which began late in the calendar year 1915 and steadily increased well into the current fiscal year. The activity is more clearly shown by the figures in Table I which cover the work up to the end of the calendar year 1916.

On July 1, 1916, there were building or under contract to be built in the United States 385 steel vessels of 1,225,784 gross tons, of which 195 of 1,037,103 gross tons were ocean steamers, each of 1,000 gross tons or over. In British yards on the same date 432 steel steamers of 1,538,640 gross tons were under construction, but the rate of progress in such construction was much reduced. The effect of this reduced output from British yards is evident when it is remembered that while at the beginning of the calendar year 1915, 1,627,316 gross tons were under construction in British yards, yet only 650,919 gross tons were launched from British yards during the year. Since late in the spring of this year, however, British shipyards have been engaged more actively in the work on unfinished merchant ships, more than doubling the output of the earlier months.

Although the total construction of merchant vessels by American shipbuilders for the fiscal year ending June 30, 1916, as reported by the Bureau of Navigation is not large, nevertheless the tonnage of ocean-going steel steamers is the largest recorded, and it is particularly to be noted that virtually half the year's product is already engaged in foreign trade. American registered steel steamers in the competitive foreign trade have increased from 1,262,159 gross tons in 1915 to 1,498,587 gross tons on June 30, 1916, a gain of nearly 20 per cent. against a gain of about 100 per cent. under the more favorable conditions of the previous fiscal year. While many vessels formerly engaged in domestic commerce have been placed in foreign trade, little has been done to replace these vessels in American domestic service, consequently it is probable that the demand for such vessels will be felt by American shipbuilders long before the present shipbuilding boom is over, and in this way the period of activity in American yards will be prolonged as the present capacity of the yards is practically booked for at least two years to come.

As matters now stand, American shipping in foreign trade has more than doubled in the last two years—a growth without precedent in the history of any maritime nation. According to the Bureau of Navigation, the increase has been in one form of employment of American ships rather than in the total shipping under the American flag. During the past two years the total American shipping has increased only 540,961 gross tons, while American shipping in foreign trade alone has increased 1,115,563 gross tons. Whether this change in the employment of American shipping will be permanent is largely a matter of conjecture, but the immediate effect of the change can only be an increased demand for

new tonnage by American shipowners for both domestic and foreign trade.—"Marine Engineering."

CO_2 RECORDERS AND THEIR VALUE.

By Victor J. Azbe.

In most boiler plants great quantities of fuel are wasted and the chief loss is that due to excess air. In many plants this loss is 18 per cent. (7 per cent. CO_2), and in some it is as high as 23 per cent. (6 per cent. CO_2). In well-operated plants the loss is ordinarily about 6 per cent. (12 per cent. CO_2), and there are some few exceptionally well designed and operated boiler plants where it is only 1.5 per cent. (16 per cent. CO_2).

To determine the loss due to excess air, flue-gas analyzers are employed, of which there are hand-operated and automatic recording types. The first ordinarily is an Orsat or one of its modifications; the second, one of the several different types of recording instruments on the market. A hand gas analyzer is a useful device, and every boiler plant, no matter how small, should have one for testing purposes. The instrument is simple, but as its operation requires considerable time and the information obtained is not immediately visible to the fireman, its value is reduced. An automatic recorder, on the other hand, shows from 10 to 20 times every hour what quantity of CO_2 is in the flue gas, and it operates continuously, thus enabling the chief engineer to know how careful or careless his firemen were during the night.

INTEREST THE FIREMEN.

It is ordinarily maintained that considerable knowledge is required to understand what the CO_2 recorder shows. This is not so. Post the CO_2 recorder conveniently for the fireman, let him know that the higher the CO_2 obtained the better fireman he is, and explain further that the higher the CO_2 the shorter will be the red line drawn by the recorder pen (or longer, depending upon the type of instrument). This is all that is necessary, except occasional suggestions on how to handle the fires to obtain a higher percentage of CO_2. In the next few weeks it will be surprising how the CO_2 will increase, especially if the firemen are placed in competition with one another by posting their results conspicuously upon a blackboard. The fireman himself will soon gain confidence, since he will see that when the fires are bad, the recorder pen strokes upon the chart will be long; when the fires are good, the ink lines will be short. It is important after he once gains this confidence that the recorders be kept in good condition and in proper operation; otherwise he will lose faith in the readings.

Improper installation, ordinarily, is the chief reason why, in so many plants, recorders are not a success. The water to the recorder must be clean, which means that any free matter must be given a chance to settle and be filtered out. The gas lines, which give the most trouble, must have the proper pitch, with no pockets, and be of the proper size. At the boiler there should be a gas filter with excelsior and cloth as filtering material, which should be changed at frequent intervals. No cocks should be allowed in the gas line; in their place oil seals should be used. Brass cocks become corroded and leak. Gas lines should be installed with the greatest care and when completed should be put under compressed-air pressure to test for tightness.

The sampling tube in the boiler uptake does not need to be of complicated design, since it is impossible to make one that will sample gas evenly throughout the whole width and length of the uptake opening. A straight tube, such as is shown in Fig. 1, is all that is necessary. The

small openings prevent the tube from becoming stopped up with soot accumulations. The position of the tube is important, and care must be exercised to have the openings placed in the center of the main gas path. If they are inserted in a dead space, lower CO_2, sometimes 3 to 4 per cent. less, will be the result.

After proper installation of the water and gas lines, the CO_2 recorder itself will give little trouble. Whatever defects arise are of a minor nature and easily remedied. The reason why so many recorders are idle or have been thrown out of the boiler plant is because the man who had them in charge did not know their principles of operation. Usually, CO_2 recorders are designed and operated upon the simplest principles of physics, and if a man knows that water will not flow uphill without some reason, that gas is lighter than water, what causes the siphon to operate and a few other such simple facts, and he takes time to learn thoroughly the duty of the different recorder parts, then he will be competent to keep the instrument in proper operation without spending on the average more than ten minutes per day per recorder.

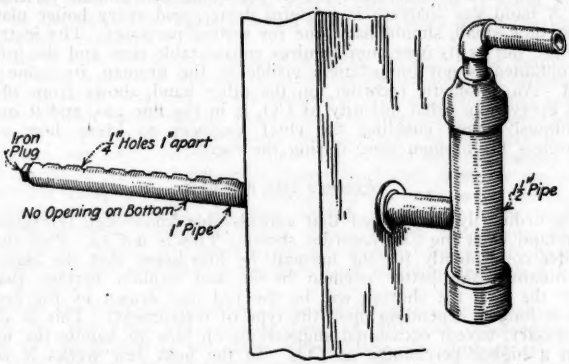


FIG. 1.—GAS-SAMPLING TUBE.

With proper adjustments, the care they require, besides changing the chart and renewing the solution, is slight. Their accuracy, with the solution changed at proper intervals, is all that is required for practical purposes. Of course, if there is the slightest leak in the gas line, the value of the recorder becomes nil, but this cannot be accepted as the fault of the recorder and can be remedied. The leak can be found by putting the line under pressure or by taking a sample of gas with a hand instrument at the boiler and comparing the result of its analysis with that of the recorder.

The cost of CO_2 recorders varies, but there is one made that sells for $180 and it is as satisfactory in operation as any other. The cost of upkeep is slight except for the caustic potash, which at the present time, owing to prevailing war prices, is quite high, requiring about $30 worth of this chemical per year per recorder. In normal times the cost is less than one-tenth of this amount.

The life of the recorder depends upon the care it receives, but it should last about five years. At the end of this time it can be overhauled for about $50 and made fit for another five years. The amount of water required varies from 250 to 500 gallons per day.

The return on the recorder investment depends upon the price of

fuel, the amount burned and the saving effected, but it can be safely accepted that hardly any other apparatus will give a greater return. In large boiler plants savings as high as $15,000 per year have been made, all going to the credit of the CO_2 recorder. In small plants the saving will be proportional and will range from 5 to 15 per cent. of the total fuel bill.

The question is often asked as to whether it is best to install a recorder for every working boiler, or only one instrument for the whole plant. The first recorder installed will produce the largest saving, since it will act as an instructor on how to fire the boilers efficiently. Having one instrument for each working boiler is of course, desirable, since with such a system all firemen are under complete check and can be put under competition. Also the boilers are operated more evenly, because with one recorder only, the fireman is liable to take good care of the boiler to which the recorder is connected and neglect the others.

As far as the boiler is concerned, an objection to the CO_2 recorder is that the fireman often tries to obtain too high CO_2. High CO_2 is accompanied by high furnace temperature, and there is a point where depreciation of the furnace lining will offset the saving effected. This point must be determined and the CO_2 percentage maintained under it. Every plant should have its own standard, which should be determined after considering all factors, the chief of which are slagging effect and incomplete combustion, both being controlled by furnace design. The standard in one plant might be 10 per cent., and in another plant of up-to-date design it might be as high as 16 or 17 per cent.

CO_2 Record — Date 5-11-16

Fireman	Boiler No. 1	2	3	4	Average
Becker	7.8	9.6	8.0		8.5%
Murphy	11.3	12.0	10		11.1%
Acton	9.5	7.3	10		8.9%

Remarks

12% CO_2 - Very Good
10% CO_2 - Fair
9% CO_2 - Bad
8% CO_2 - Very Bad

FIG. 2.—BLACKBOARD CO_2 RECORD.

Fig. 2 shows a blackboard design on which to post firemen's CO_2 percentages. This record system can be elaborated, and at the same time complicated, but the form shown will be found simple and effective. When greater accuracy is desired, it would be advisable to post the corresponding fuel loss instead of the CO_2 percentage, as the fuel loss is not directly proportional. For example, raising the CO_2 from 8 to 9 per cent. will cause a fuel reduction of 3.3 per cent. while raising it from 11 to 12 per cent. the saving will be only 1.8 per cent. The loss could also be expressed in percentage considering the maximum CO_2, or the standard for the plant as 100 per cent., the minimum allowable as 0 and determining the intermediate points from the fuel-loss table. But there is danger that if the system is made too complicated for the fireman to grasp, he will lose interest, and of course this is what is not wanted.

Herewith is presented a table showing the fuel loss at different CO_2 percentages, based upon a flue temperature of 600 degrees F. For any other temperature it will be necessary to multiply by the correction factor given.

FUEL LOSS DUE TO EXCESS AIR.

CO_2, %	Pure Carbon OH, %	Bituminous Coal, 3%H, %	Fuel Oil, 10%H, %	Natural Gas, 70%H, %
21	0	....	....	....
20	0.2	....	....	....
19	0.7	0.2	....	....
18	1.6	0.8	....	....
17	2.7	1.6	....	....
16	3.3	2.5	0.2	....
15	4.6	3.5	1.0	....
14	5.9	4.5	2.0	....
13	7.2	5.6	3.2	....
12	8.5	7.1	4.6	....
11	10.2	9.0	6.0	....
10	12.5	11.3	7.5	....
9	15.5	13.7	9.6	....
8	18.7	16.9	12.5	....
7	23.7	21.3	15.6	....
6	29.3	26.2	20.0	....
5	36.7	30.5	26.2	....
4	49.3	45.0	34.3	....
3	68.7	62.5	50.0	1.5
2	....	....	78.7	7.1
1	....	....	....	22.5
0.5	....	....	....	50.0

CORRECTION FACTOR.

Temperature.....	450	500	550	600	650	700	750
Factor	0.7	0.8	0.9	1.0	1.1	1.2	1.3

Gas analysis is considered by the best engineers as the most convenient and effective method of checking boiler-room performance. It is something that will stay with us in the future and will be more generally used than at present. Such statement by engineers as, "We don't want CO_2, we want steam," is ridiculous. An erroneous impression prevails that to maintain high CO_2 it is necessary to close the damper. Of course it is more easy to obtain high CO_2 with low draft than with high, nevertheless as high a percentage can be maintained with the latter condition if the fire is in the proper condition. As far as steam is concerned, there will be more of it generated with a given draft and high CO_2 than with low CO_2.—"Power."

FORCED LUBRICATIONS FOR MARINE TURBINES.

With the more or less general adoption of geared turbines to all classes of ocean-going vessels, in most of which, unlike the navy, there are only two engineers on watch, it has become necessary to give the subject of lubrication considerable thought with a view to obtaining a degree of simplicity and dependability which will reduce attention and anxiety to a minimum.

When considering the adoption of any particular system, it must be borne in mind that no hitch should be possible which might in any way make it necessary to stop the turbines, otherwise, under difficult sea conditions, results might be disastrous. To meet such a condition, all important fittings which in themselves could obstruct the passage of oil

must be supplied in duplicate, or be capable of being by-passed and be easy of rapid overhaul.

As every trade route has its own conditions to meet, it is impossible to describe a system suitable for every class of ship, but it is proposed to describe a system which will meet the requirements of many single-screw cargo-carrying vessels, and which can be modified to meet special conditions, arising from long voyages, tropical waters, &c.

A combined gravity and forced system offers the greatest advantages, and consists simply of a pump taking oil from a drain tank and discharging it through a cooler to a gravity tank, from which it falls to the turbine bearings, &c., and back to the drain tank. Or can be by-passed, in which case it does not enter the gravity tank, but is forced direct to the machinery.

The following main fittings would be required for such a system, and are stated in the order in which the oil would pass through them, when travelling from the drain tank to the turbines:—One drain tank; two oil strainers (one for each pump); two pumps (one working and one stand-by); one oil cooler; two gravity tanks; and one combined oil distributor and water and sludge collector. There are many small fittings of a more or less essential character, and these will be dealt with in describing the above main fittings.

The drain tank should be of sufficient capacity. when full, to supply oil for fifteen minutes, assuming none is returning to the drain tank, or, in other words, the oil should be travelling four complete circuits in an hour. The tank should be a self-contained unit placed in a cofferdam in the double bottom rather than built into, and forming part of, the double bottom, and should be easy of removal for examination. The top should be not less than 6 inches below bottom of the gear case to ensure a ready drain, and not less than 3 inches above double-bottom tank, to clear bilge water. In addition to the drain and suction pipe connections, a float level indicator must be fitted so as to be easily observed from the starting platform, also an air pipe and manhole.

The oil strainers perform an important duty and deserve attention in design. The body may be of cast-iron, having a cover easy of removal, to enable inspection and cleaning of the straining grid. The grid may be a gunmetal casting or a steel tube perforated with holes, say 1 inch diameter, and in either case the area through grid should be about 10 times that of suction pipe. A suitable straining medium will be found in 24-mesh wire gauze, this to be properly secured to the grid. The gauze will reduce the effective area through grid by about half. The strainer should be in an accessible position, and have the cover on top to avoid overflow of oil when cleaning.

The pumps are made by specialists and need no particular description; however, they should be ordered of ample capacity and be fitted with a valve or cock on discharge, and suction branches for isolation purposes in case of overhaul at sea. For guidance purposes the speed of oil through discharge pipes may be taken at 200 feet per minute for pipes 3 inches diameter and above.

The oil cooler calls for special attention, particularly when the vessel passes through tropical waters. It may take the same form as a steam condenser, with similar tubes and packing, but with the water circulating around outside of tubes, with preferably a four or six flow to ensure good circulation. This plan has been largely adopted, but it is economically impossible to render the flow of the oil turbulent, and the rate of heat transfer is accordingly low. Even viscous fluids, however, will not maintain the laminar character of the flow around a sharp corner. Hence certain firms construct their coolers in a different way. The water is passed through the tubes and the oil outside. Numerous baffles are fitted to cause the oil to flow up and down in zigzag fashion across the cooling tubes, and at each abrupt change of direction a certain disturbance is

produced in the laminar flow of the oil which facilitates the interchange of heat. At present knowledge of oil coolers is in its infancy, and there is much room for research and careful design. It is desirable to fit thermometer pockets on oil and water inlet and outlet. The present practice admits of oil discharging to turbines at 80 degrees F., and leaving turbines at a maximum of 120 degrees F. The cooler should be easily accessible and have a by-pass fitted.

Each gravity tank should be of equal capacity to the drain tank and be fitted with cocks to enable either tank to be in circulating use while the other acts as a settling tank. These tanks must be placed high enough to give a gravity pressure of at least 10 pounds per square inch, and should be fitted with gauge glass and float indicator with wire leads and gauge board arranged to enable the level to be observed from the starting platform. A manhole, air pipe and overflow pipe must be fitted, the overflow pipe being led back to the drain tank or reserve oil tanks. As previously mentioned, a by-pass is fitted between the pump and the gravity tank to enable forced lubrication to be applied to the turbines without the oil having to pass through the gravity tanks.

The combined oil lubricator and water and sludge collector need be nothing more than a cylindrical casting of sufficient depth to admit of water and sludge gathering at the bottom, while oil flows out at the top to the bearings and gear sprayers. It is desirable to fit separate connections from the distributor to the bearings and gear sprayers, with a controlling valve at each, as an adjustment is thereby easily effected whereby the bearings and sprayers will each receive their proper share of oil. A water draw-off cock placed about 12 inches from the bottom and a door for removal of sludge are to be provided.

Having covered the main fittings, a word or two on the turbine fittings might not be out of place. Control valves on each turbine bearing are unnecessary, provided the pipe arrangements are such that all bearings get an equal chance. To ensure this, pipes should have easy bends and branches on pieces be given direction with flow of oil. Means of testing the flow to bearings must be provided, and may take the form of a sight glass placed in the line of pipe, or of a test cock on the high-pressure side of a bearing, having a visible flow to the drain side. As an extra precaution pressure gauges may be fitted on the pressure side. The gearing sprayers should each be fitted with a guard cock having a locked-open arrangement.

The pipes and fittings should be tested to 50 pounds per square inch, with the exception of the tanks, for which a 5-inch test is sufficient. A relief valve should be provided near the pump set to blow-off at 50 pounds per square inch, having an outlet led to the pump suction. The pipes may be of steam-quality iron, with the exception of the small sizes, where it will be found cheaper and more convenient to fit copper pipes with nipples. Great care should be taken in shops to clean out turbine, gear case and other castings thoroughly. A steam jet will be found a convenient means of removing all traces of sand and other foreign matter, while linen makes a good finishing-off cloth. Cotton or other waste should not be used, as much trouble has sometimes arisen due to the strands of thread, &c., left on the castings. It is impossible to lay too much stress on the importance of this necessity for thorough cleaning

It will be noted that nothing has been said regarding the water circulation to the oil cooler; this can be arranged as desired, but a separate pump is undoubtedly best, with a stand-by connection from main condenser circulating pump. If this latter is fitted, a good circulation is obtained by leading the discharge from the cooler to the main circulating pump suction. Should a separate overboard valve be fitted, it must not be placed higher on the ship's side than the main circulating discharge; otherwise the heads are unequal and the cooler might suffer accordingly.

The system described is a simple one, with as few complications as

possible, and can be adapted to most vessels. It has been pointed out in some quarters that particles of metal from the gearing mix with the oil and gradually work their way into the bearings, with bad effect. To avoid this, separate systems have been fitted, one to the turbine bearings, and the other to the sprayers and gearing bearings, it being necessary to combine the two latter, due to the drain being a common one into the gear case. It is very doubtful if such a partial correction is worth all the duplication of working parts necessary to carry it out, and it is also doubtful if with good quality of gearing and reasonable changes of oil any trouble may be expected from the bearings.—"Engineering."

ALQUIST GEARING FOR SHIP PROPULSION.*

By W. L. R. Emmet, Member of Council.

The designs described in this paper are based upon the inventions of Mr. Karl Alquist. Mr. Alquist is an accomplished engineer and was formerly connected with the turbine department of the English branch of the General Electric Company, the British Thomson-Houston Company. His gear inventions were first brought to the attention of the writer early in the year 1911. For some time previous Mr. Alquist had endeavored to arouse interest in his methods in England and on the Continent, but had accomplished nothing. At that time the General Electric Company had not begun the commercial manufacture of high-speed spiral gears, but had for some time been conducting experiments to determine the limits of speed, pressure, &c., which were practicable with such gearing. The importance of high-speed gearing in connection with turbine and electrical applications is obvious and we were working with a view to development of the best standards.

Certain features of Mr. Alquist's proposals appealed strongly to the writer, and arrangements were made with Mr. Alquist to come to America and assist in experimental developments along the line of his inventions.

The result of this undertaking has been that many sets of gearing have been built and experimented with exhaustively under a variety of conditions, and by these experiments certain standards of practicability have been established and extensive commercial developments have been undertaken. Gearing of this character has been applied to about 72 sets where steam turbines drive electric generators of various types. Contracts have been closed for machinery for the propulsion of 70 ships aggregating 215,200 h.p. Some of these electric generating sets have been in service 1½ years, and about seven of the ship sets are in service, some of them having made many long voyages. Among these are high-pressure cruising units for the battleship *Nevada*, which have been in service for some time and shown very fine results. Among the ship equipments not yet completed are included the propelling machinery for Destroyer *No. 69*, built at Mare Island, and new propelling machinery for the scout cruiser *Salem*. In all of this practical experience no case of trouble with gearing has developed and no appreciable deterioration of gears has been observed.

One of the important reasons for adopting this type of gearing was that its design tended to afford a distribution of strains and means by which excessive strains would not be imposed upon any part through slight imperfections, distortions, or inaccuracies. The uniform success which has been accomplished with an entirely new product shows that this expecta-

*Paper read before the American Society of Naval Architects and Marine Engineers, at New York, November, 1916.

tion has been amply justified. Some of the gears which have been used have been very imperfect, both in the matter of material and workmanship, and have been used under extremely trying conditions. That they have not failed has afforded the strongest evidence of the general reliability of the method.

The character of construction used in this gearing is shown by one of the drawings, Fig. 1. The gear is built up of a number of plates machined

Fig.1.

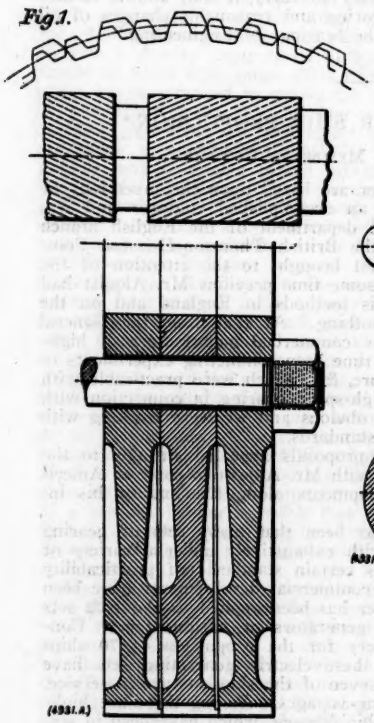
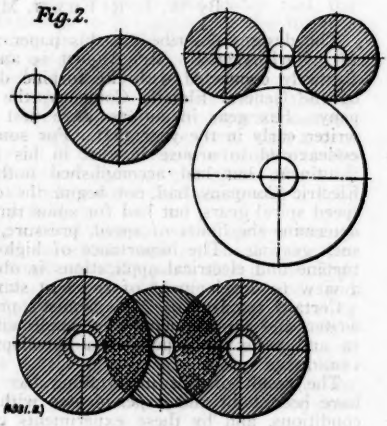


Fig.2.



to a form which gives them the desired degree of lateral flexibility. These plates are put together, engaging solidly at the hub, and also engaging on a narrow edge at the periphery. When so built together they form a solid cylinder which can be spirally cut in the ordinary manner. After cutting, the edge engagements are relieved with a small dividing tool, so that each disc operates independently and is free to deflect laterally under the side pressure which results from its diagonal engagement with the pinion. The parts are so proportioned that this lateral deflection can at no time involve fibre strains which could possibly cause destructive fatigue. A very small amount of this lateral deflection is sufficient to afford the desired distribution of load, and this amount can easily be given without approaching dangerous periodic strains.

To appreciate the value of this flexibility and load distribution, various

peculiarities of solid and spiral gearing must be considered. In the first place, where gears are inflexible there must always be a tendency to increase strain at the loaded end of the pinion through torsional deflection of the pinion. There is also a tendency to inequality of strain on different parts of the surface through the lateral deflection of the pinion under load. These inequalities can be partially compensated by elevating the bearings or evening the pressure on the bearings, but this compensation can only be partial because the correction applies only to the two ends and not to the middle. Furthermore, the momentary and periodic strains on different points of solid spiral gearing may be seriously affected by vibrations of supporting structures, irregularities of machine work or gear cutting, and other causes. If for any reason such conditions cause any tooth or part of a tooth to receive periodically excessive strains, fatigue may result and a broken tooth may destroy the whole gear. To obviate the possibility of breakage under such conditions it is often desirable in solid gearing to use relatively large teeth in order that these possible irregular strains on teeth or parts of teeth will not involve danger of breakage. For other reasons, however, the use of large teeth is distinctly undesirable in spiral gearing. Spiral gears tend to engage by point contacts at or near the pitch line, and the ability of these point contacts to bear pressure without fatigue of the surface metal is governed largely by the flatness of the surfaces engaging rather than by the size of the teeth carrying these surfaces. The flatness of the surface is a function of the pinion diameter and not of the pitch. If we double the number of teeth in a spiral gear we have twice the number of driving points in action, and the flatness of all of these points is the same in both cases if the pitch diameters are the same. These matters are illustrated in the sketches in Fig. 1.

With gearing of the Alquist type we can use very small teeth without any danger of incurring excessive strains on individual teeth, which might involve risk of the development of fatigue cracks. In this connection it should be borne in mind that experiments have shown that the strongest steel, if subjected to periodic deflections, will break after a fiber strain of 20,000 pounds per square inch has been applied 1,000,000 or more times.

In the work which is now being done by the General Electric Company, gears of the type described are applied in three ways. First, a single reduction has been accomplished by engaging one solid pinion with a flexible gear of this type; second, by engaging a solid pinion with two flexible idlers, which idlers in turn engage with a solid large gear; and third, in a double reduction where a solid, high-speed pinion engages flexible gears on two countershafts, these countershafts carrying solid pinions, both of which engage a flexible gear on the same low-speed shaft. In these two latter applications the flexibility of the gears serves to equalize the loads between all of the driving points, and the use of a plurality of driving points on the large gear reduces the length of face necessary on that gear. These different methods of application are illustrated by diagrams in Fig. 2, and are also shown by the photographs and drawings of actual machines, Figs 3 to 8.

In both of these cases where a single high-speed pinion drives two flexible gears, other very positive advantages are accomplished. The pinion is relieved of bending strains, and pressure on the high-speed bearings is avoided.

Experiments have been made in Schenectady which carefully analyze the losses of high-speed gears under various conditions of load and pressure, and these experiments have indicated that low peripheral speeds are more efficient than high speeds. With solid gears low speeds involve difficulties because they diminish pinion diameters and increase pinion lengths, thus complicating the troubles which may arise through deflection and torsion of pinions and pressure upon pinion bearings. Since the flexibility of Alquist gears enables us to compensate easily for all possible degrees of torsional yield in the pinion, we can with such gears safely

reduce peripheral speeds by making the gears longer and of smaller diameter. Such reductions of diameter diminish weight and improve efficiency, and the fact that the Alquist method makes such reductions possible constitutes one of its important advantages.

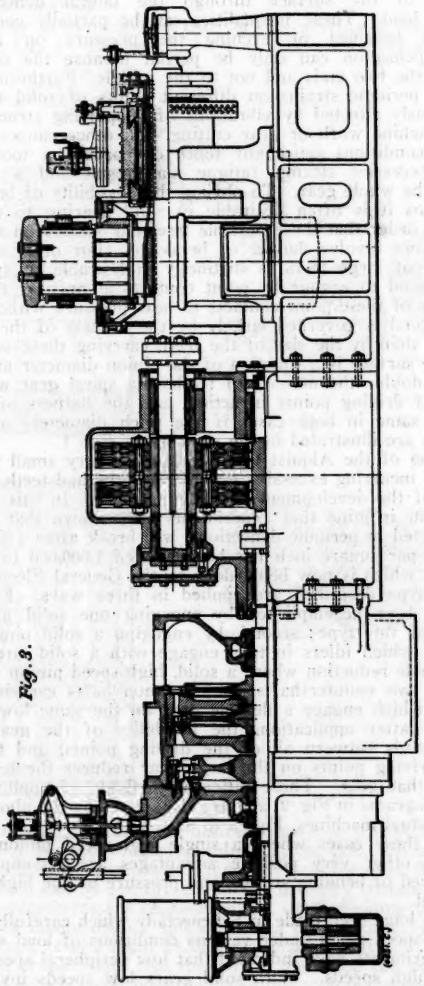
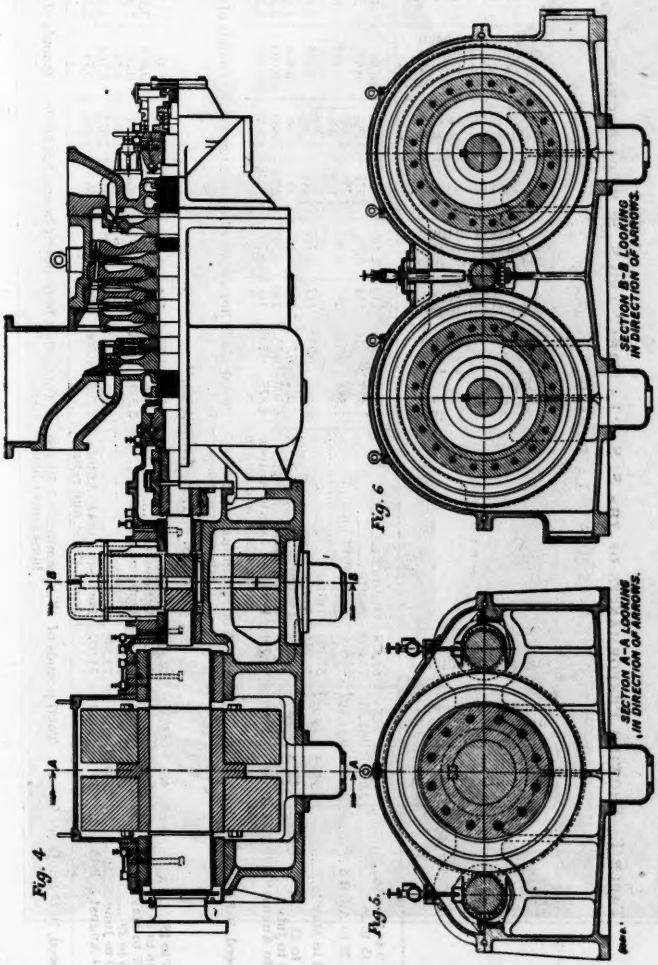


Fig. 3.

Table I, annexed, shows a comparison of performances for two sister ships, the *La Brea* and the *Los Angeles*, operated by the Union Oil Company of Los Angeles. Both of these ships are new, carry similar car-

goes, and burn the same fuel oil in the same boilers under similar conditions. The *La Brea* is fitted with our geared-turbine equipment, and the *Los Angeles* is fitted with triple-expansion engines of the best type. The *Los Angeles* is equipped with steam-driven oil-handling pumps of the



ordinary type, and the *La Brea* is fitted with a new arrangement of pumps driven from the upper deck by General Electric motors. The table illustrates the superiority of the oil-handling machinery as well as that which propels the ships. The horse-power hours given in this table are calculated from the propeller and hull data given by the naval architect

TABLE I.—SHOWING PERFORMANCES OF THE S. S. "LA BREA" AND S. S. "LOS ANGELES."

S. S. "LA BREA."

Voyage.	Date.	Total distance in knots.	Average speed in knots per hour.	Total barrels delivered.	Name of port.	Total fuel used steaming on voyage, in barrels.	Total barrels per knot, steaming.	Total barrels used in port.	Total barrels per knot steaming and in port.	Time in port, hours.	Barrels discharged per hour.	Cost at 80 cents per barrel per knot, steaming and in port (fuel).	Shaft horsepower-hours.	Pounds of oil per shaft horsepower.
1	March 9 to 15, 1916	650	9.5	62,578	Oleum	574	0.883	142	1.10	36½	1,714	$0.88	116,000	1.67
2	March 15 to 26	2,037	11.33	73,600	Seattle	1,459	0.716	152	0.79	37½	1,962	0.632	505,000	0.941
3	March 28 to April 6	2,108	11.01	64,676 8,642	Vancouver Seattle	1,584	0.751	155	0.824	26 13½	2,487 640	0.659	530,000	1.01
4	April 8 to May 16	9,254.5	10.97	52,109 19,045	Taltal Antofagasta	6,896	0.745	117	0.757	33½ 15½	1,544 1,228	0.605	2,330,000	1.00
5	May 7 to 23	450	11.20	77,292	Oleum	321	0.713	134	1.01	39	1,980	0.808	1,180,000	0.911
6	May 24 to July 1	9,196.5	10.65	71,824	Antofagasta	6,900	0.75	109	0.762	25	2,660	0.609	2,200,000	1.06
7	July 1 to August 9, 1916	9,184	10.39	71,791	Port San Luis to Antofagasta, Chile, and return.	6,875	0.748	107	0.76	42.3	1,697	0.609	2,230,000	1.04

Average speed, 10.7 knots per hour. Total pounds of oil, steaming = 8,270,000. Total shaft horsepower-hours, 8,029,000. Pounds of oil per shaft horsepower-hour = 1.03.

S. S. "LOS ANGELES."

1	April 9 to 15	423	9.4	67,674	Oleum	556	1.31	164	1.70	64½	1,099	1.36	74,600	2.48
2	April 16 to 25	1,845	10.13	74,739	Vancouver	1,656	0.897	218	1.01	33½	2,214	0.808	3,940,000	1.41
3	April 27 to May 25	6,549	10.22	73,734	Panama	5,579	0.851	169	0.877	46½	1,577	0.701	1,420,000	1.315
4	May 26 to 29	220	9.1	72,372	Oleum	221	1.00	123	1.56	41	1,770	1.24	39,000	1.88
5	May 30 to June 27	6,348	10.6	72,538	Balboa	5,462	0.86	159	0.885	35	2,072	0.708	1,460,000	1.255
6	June 29 August 7, 1916	9,151	10.24	71,007	Port San Luis to Antofagasta, Chile, and return.	8,293	0.906	186	0.926	56.3	1,261	0.74	2,130,000	1.31

Average speed, 10.27 knots per hour. Total pounds of oil, steaming = 7,310,000. Total shaft horsepower-hours, 5,538,000. Pounds of oil per shaft horsepower-hour = 1.320.

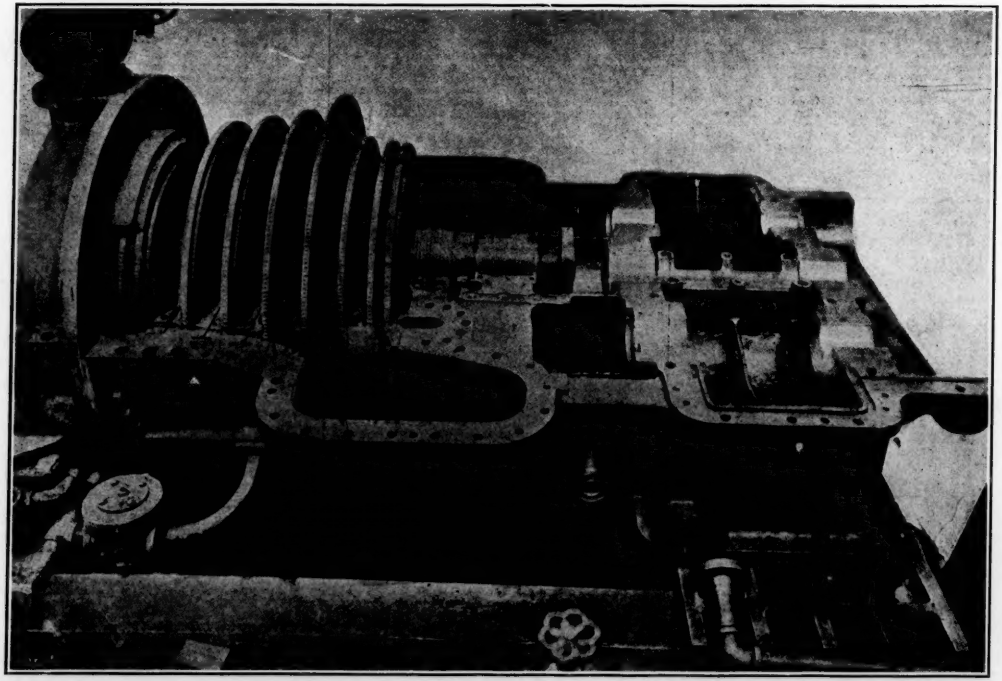


FIG. 7.—ALQUIST GEARING FOR STEAM TURBINES.

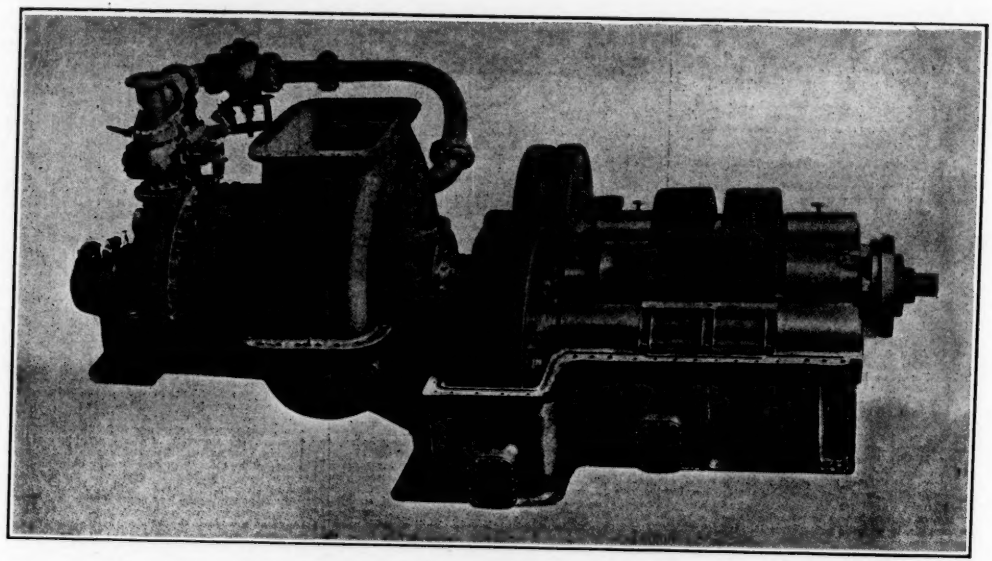


FIG. 8.—ALQUIST GEARING FOR STEAM TURBINES.

of the Union Iron Works who built the ships, and based upon model tank experiments which were made in Washington. The models of the ships are identical, the *La Brea* propeller operating at 90 r.p.m. and that of the *Los Angeles* at 65 r.p.m.

The seventh voyage of the *La Brea* and the sixth voyage of the *Los Angeles* were made over the same course and at nearly the same time. The relation of fuel economy of these two voyages almost exactly corresponds to that shown by the average of all the voyages and tends to verify the accuracy of the whole comparison.—"Engineering."

BOOKS RECEIVED.

STEAM TURBINES, by C. J. MEYERS, Lieutenant Commander, U. S. Navy, U. S. Naval Institute, Annapolis, Md. Price, $4.50 (subject to change).

This book covers turbine installations in the U. S. Navy with chapters on the elementary principles of design and construction, and instructions for care and operation. Especially adapted for use as an elementary text book because it contains in one volume descriptions and excellent illustrations of various types of turbines, and does not go too deeply into the theory and principles of design. Illustrated with 165 text figures and 9 plates 11 × 40 inches, showing half cross section of the latest Curtis and Parsons turbines.

STRENGTH OF SHIPS, by A. J. MURRAY, Longmans, Green & Co., Fourth Ave. and 30th St., New York City; 39 Paternoster Row, London. Price, $5.00 net.

A book that thoroughly covers the materials used in ship construction and the principles of design for strength of the component parts of the ships, of the ship as a whole, and of the rigging and outboard fittings. The book is well printed and illustrated, and should be of value as a reference book and a text book for advanced students.

ASSOCIATION NOTES.

A meeting of the Society was held on Friday, December 29, 1916, for the purpose of counting ballots cast for officers of the Society for the year 1917, and for and against the holding of a banquet in that year.

The following were declared elected officers of the Society for 1917:

President, Captain H. P. Norton, U. S. Navy.

Secretary-Treasurer, Lieutenant Commander J. O. Richardson, U. S. Navy.

Council:

Captain T. W. Kinkaid, U. S. Navy.
Naval Constructor Robert Stocker, U. S. Navy.
1st Lieutenant of Engineers Q. B. Newman, U. S. C. G.

The vote on holding a banquet resulted as follows:

For, ...	224
Against, ...	80

The following were appointed a committee to arrange for a banquet, with full power to act:

Commander R. H. Leigh, U. S. Navy;
Lieutenant Commander J. H. Tomb, U. S. Navy;
Lieutenant Commander G. W. S. Castle, U. S. Navy;
Lieutenant Commander J. O. Richardson, U. S. Navy.

The committee made arrangements for holding the banquet at the Army and Navy Club, Washington, D. C., on Saturday, February 24, 1917. In view of the uncertain conditions in the service recently arising, however, the Council of the Society, with great reluctance, decided that the banquet should not be held, and all arrangements were canceled.

The following members and associates have joined the Society since the publication of the last number of the JOURNAL:

MEMBERS.

Ducey, David F., Lieutenant, U. S. Navy.
Elder, Fred K., Lieutenant, U. S. Navy.
Lind, Wallace L., Lieutenant, U. S. Navy.
Ziegemeir, Henry J., Commander, U. S. Navy.

ASSOCIATES.

Beckett, George W., Lieutenant, S. C. N. M., Beaufort, S. C.
Braun, Carl F., President, C. F. Braun & Co., 503 Market St., San Francisco, Cal.
Eshbach, Ovid W., Dept. of Electrical Engineering, Lehigh University, South Bethlehem, Pa.
Isaacs, Landis, Ensign, P. A. N. M., 612 West 10th St., Erie, Pa.

FINANCIAL STATEMENT, 1916.

Publication Statement.

The following statement shows the returns from and costs of the issue of the JOURNAL of the Society for the year 1916:

RETURNS.

Dues	$5,418.78
Subscriptions	2,100.20
Sales	315.50
Advertisements	2,431.25
Exchange	2.70
	$10,268.43

COSTS.

Printing	$5,558.84	
Engraving	742.20	
Drafting	19.00	
Manuscript	1,280.70	
Postage and expressage	157.93	
Commission	161.07	$7,919.74
Profit on issue of JOURNAL		$2,348.69

GENERAL PROFIT AND LOSS STATEMENT.

The following statement shows the result of the business as a whole during the year 1916:

	Loss.	*Profit.*
Profit on issue of JOURNAL		$2,348.69
Interest		387.22
		$2,735.91
Banquet, 1916	$461.59	
General expense	1,610.08	
Current profit and loss account	28.90	2,100.57
Net profit		$635.34

BALANCE STATEMENT.

The following statement shows the difference between the total assets on January 1, 1916, and those on December 30, 1916:

Assets December 30, 1916.

Cash	$3,019.12
Accounts receivable, advertisements	1,346.38
Accounts receivable, dues	1,080.00
Accounts receivable, sales	127.96
Accounts receivable, subscriptions	503.52
1 Army and Navy Club bond (cost)	1,000.00
6 Washington Railway and Electric bonds (cost)	5,215.50
Furniture	62.00
Miscellaneous asset (gold medal)	16.00
Total	$12,370.48
Total assets January 1, 1916	11,735.14
Net gain	$635.34

NOTE.—Market value of Washington Railway and Electric bonds on December 31, 1916, was 82. We paid 81 for one and 88 for five. They have always been carried at cost in all previous statements.

Army and Navy Club bond is not a listed bond.

The numerical strength of the Society on December 31, 1916, was—

Active members,	724
Associate members,	395
Subscribers,	394
Exchanges,	77
Total,	1,590

Audited and found correct.

J. O. RICHARDSON,
W. A. SMEAD,
ALEXANDER SHARP, JR.

A member of the Society offers for sale a set of the JOURNAL, Vols. I to XXVIII, inclusive, except No. 3, Vol. XVIII, and No. 2, Vol. XX, with indices. The first five volumes bound in cloth. In good condition. Inquiries should be addressed V, care of American Society of Naval Engineers, Navy Department, Washington, D. C.

J. O. RICHARDSON.